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The Proficorder—An Instrument for Recording Waviness and Other Surface Profiles

By E. J. ABBOTT¹ AND EDWARD RUPKE,² ANN ARBOR, MICH.

The Proficorder described in this paper represents a major addition to available surface-measuring instruments. It provides data which are highly essential in the study of surfaces and the processes by which they are obtained. Its function is to reproduce, with appropriate magnifications, the actual profile of a considerable length of surface. It permits the study of individual irregularities as to size, shape, and relative position with respect to other irregularities. The instrument can be used to good advantage for inspection of production setups for waves and other surface characteristics. As an instrument for use in process engineering, it provides quantitative data for determining the effectiveness of steps taken to improve or obtain certain surface finishes in machining or grinding operations. As a laboratory instrument, the Proficorder supplies indispensable information to all those who are interested in determining the causes, effects, and characteristics of surface irregularities.

IN the last two decades industry has become much concerned with surface finish, particularly with the types of surfaces produced by various machining and grinding operations. Considerable progress has been achieved in the development of shop instruments for the measurement of surface roughness, i.e., irregularities having a horizontal spacing of the order of thousandths of an inch. Until now, however, there has been a complete lack of instrumentation for shop use which could record the size and shape of more widely spaced irregularities, such as, waves, steps, or bows. The horizontal widths of these irregularities fall outside the scope of the available instruments used in measuring surface roughness.

The Proficorder was developed primarily to fill this gap. It provides means for recording surface irregularities which have horizontal spacings up to an inch or more, and which thus fall in the category of waviness. Waviness is an inherent characteristic of all machined and ground surfaces, and is an undesirable feature in many applications. Accordingly, there is a definite practical need for a shop instrument to measure these waves, and other widely spaced irregularities, so they can be held within acceptable limits.

The necessity for a shop instrument, in contrast to a laboratory type, arises from the fact that the waviness of machined and ground surfaces is the result of a complex and subtle combination of factors in machine setup. These factors include the following:

- 1 Condition of cutting edges of tools and grits of wheels.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

- 2 Selection of feeds, speeds, coolants, etc.

- 3 Condition of bearings, spindles, etc.

The cutting edges of tools and the grits of wheels are the primary factors in the production of waviness. They are also the most difficult factors to specify and control. Accordingly, the only practical test for the acceptability of a machining or grinding setup is the acceptability of the finished part. In making any machine setup, therefore, it is necessary to check the first few parts produced in order to determine whether or not they are within the desired tolerances. The Proficorder provides the first direct means for measuring the waviness of such pieces in order to check a setup for its initial acceptability as to waviness. Furthermore, because the quality of the cutting edges of tools and grits of wheels may change significantly during an ordinary run, it is desirable to repeat this check at intervals during a run. This procedure of measuring the waviness of the parts produced also provides the most practical method of checking the selection of speeds, feeds, coolants, etc., as well as showing up defective conditions of bearings, spindles, etc. For making these initial and periodic checks it is essential that the measuring instrument be conveniently available at all times to the machine where the work is being done.

For some years it has been possible to measure waviness by means of laboratory instruments and techniques, but this method has served only to provide experimental data and not as a means for controlling machine setups in various production processes. The Proficorder was, therefore, developed to make on-the-spot checks of parts and surfaces as they are produced by a particular machine. In order to serve in this capacity, it has the following characteristics which are essential to all satisfactory shop instruments:

- 1 Simple and easy to operate.
- 2 Rugged and reliable over long periods of time under shop conditions.
- 3 Versatility in accommodating the various parts and types of surfaces that are machined and ground.

The present instrument can record surface profiles over a distance of approximately 2 in. The record obtained represents the actual contour of the surface traced. It shows the finely spaced irregularities, or roughness, superimposed on the more widely spaced irregularities, such as waves or bows. Steps, flaws, or other nonrecurrent irregularities are likewise registered, showing their true relationship to the rest of the surface.

BASIC DESCRIPTION OF INSTRUMENT

The focal point of the Proficorder is the chart record of the profile of the surface in question. The record is drawn in ink by the pen element of a recording meter. The recording of a usable profile on this chart entails the following necessary requirements:

- 1 Vertical magnification.
- 2 Horizontal magnification.
- 3 A tracing mechanism.

Vertical Magnification. The vertical magnitude of surface irregularities is usually of the order of microinches. Accordingly, this dimension must be greatly magnified in order to obtain a usable chart for depicting such small irregularities. The magnification must be accurately calibrated in order to determine the actual height of the irregularities in question.

Horizontal Magnification. Adequate or appropriate horizontal spacing between the irregularities on a profile record is necessary in order to identify the individual irregularities, or groups of irregularities. The line drawn by the pen is wider than the actual spacing between most of the finer irregularities on the surface. Therefore it is essential that a means be provided for stretching out the record of the surface as it is being recorded on the chart. The amount of magnification desired is dependent upon the character of the surface being profiled and upon the type of irregularities which one is interested in studying.

Tracing Mechanism. The profile record on the chart is the end result of the displacements of a tracer point as it moves across the surface being profiled. The means and accuracy by which this tracer point is moved over the surface is highly important with respect to the validity of the profile record. A satisfactory tracing mechanism must have both accuracy and adaptability to a wide variety of surfaces.

PHYSICAL COMPONENTS OF INSTRUMENT

The essential features listed in the previous section are incorporated in three distinct physical units, i.e., tracer, piloting fixture, and amplicorder. Fig. 1 shows a typical arrangement.

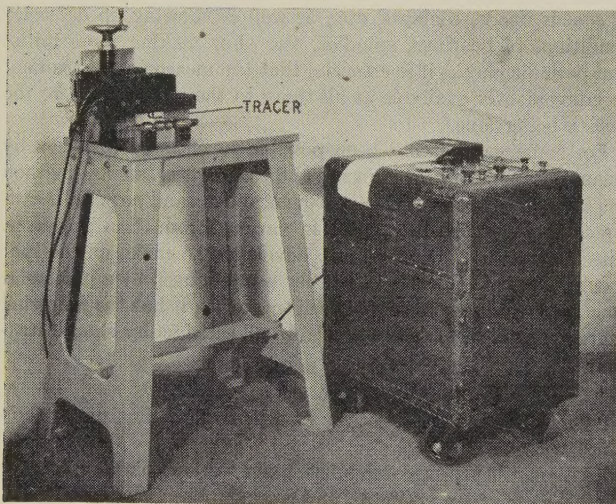


FIG. 1 THE PROFICORDER SETUP

(Type RLA piloting fixture at left; Amplicorder unit at right. Specimen mounted on table of fixture and tracer directly above.)

Each of these three basic units is a logical integral assembly by itself, but functionally they are interrelated in a rather complex manner. The essential features and functions of each are as follows:

Tracer

- (a) A tracer point which follows the surface being profiled.
- (b) A conversion element for changing mechanical displacements into electrical voltages.

Piloting Fixture

- (a) Means for mounting the tracer and moving it in an optically flat plane over approximately 2 in. of trace.
- (b) A tilting adjustment for aligning the tracer with the specimen.

- (c) Vertical adjustment to accommodate different sized specimens and for mounting or removing the specimens.

- (d) A means for mounting and supporting the specimen being measured.

Amplicorder

- (a) Oscillator to provide "carrier" for measuring circuit.
- (b) Amplifier.
- (c) Demodulator.
- (d) Recording meter.
- (e) Chart.
- (f) "Range switch" for selecting desired vertical magnification.
- (g) Controls for horizontal magnification.
- (h) Driving mechanism for chart and piloting fixture.
- (i) Provision for manual operation of piloting-fixture drive.
- (j) Drive and adjustment for tracing speed.
- (k) Secondary controls and switches for convenience of operation.

PRINCIPLES OF OPERATION AND FUNCTIONAL DETAILS

As indicated earlier, the three basic requirements for successful profile recording are vertical magnification, horizontal magnification, and a tracing mechanism. The following description of the Proficorder shows how these basic requirements have been incorporated in the physical units listed in the previous section.

Vertical Magnification. The magnification of the mechanical displacement of the tracer point is obtained by means of an electronic measuring circuit. The essential feature of this measuring circuit is that it records static, as well as dynamic displacements of the tracer point. In recording waves which have wide horizontal spacing, or steps which have a single nonoscillatory displacement, it is necessary that the measuring circuit be able to record nonrepetitive or "zero-frequency" displacements of the tracer point.

When using the tracer method, there are two principal means for producing voltages:

- (a) Generation of a voltage by means of the mechanical oscillations of the tracer point.
- (b) Modulation of a carrier voltage.

(a) In the generating method, the frequency of the electrical voltage produced is the same as the frequency of the mechanical oscillations of the tracer point. Therefore it follows that at zero frequency of the tracer point, the voltage output of the tracer is zero. A moving coil in a magnetic field and a piezo crystal are examples of the generating method of producing voltages. In order to obtain measurements, this generating type of mechanical-electrical conversion must be operated above some minimum "cutoff" frequency. When operated near or below this minimum frequency, serious discrepancies appear in the conversion. Because of this low-frequency cutoff, therefore, it becomes impossible to use the generating-type circuit to show the shape and dimensions of the irregularities with larger horizontal spacing, which fall under the classification of waviness. Moreover, steps or nonrecurrent irregularities of small horizontal dimension will produce only intermittent voltages, and thus will not show their shape with respect to the rest of the surface. For these reasons, the generating system of producing voltages has serious limitations. With this type of measuring circuit, it is impossible to make profile records that will show the size and shape of waves or nonrecurrent irregularities.

(b) The modulated carrier method of producing voltages involves complexity of design, but offers the only feasible method

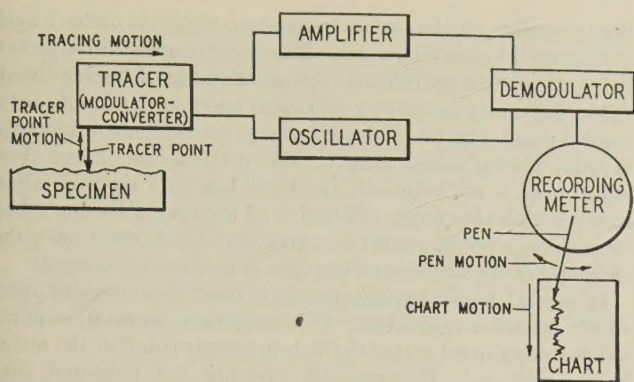


FIG. 2 BLOCK DIAGRAM FOR PROFICORDER MEASURING CIRCUIT

for including zero frequency in the measurement band. As explained previously, this inclusion is a necessity for measuring waviness. Therefore the modulation type of mechanical-electrical conversion has been used in the Proficorder.

Fig. 2 is a block diagram of the various components of this circuit for recording static and dynamic displacements. The oscillator is the vacuum-tube type, and provides a carrier voltage of constant frequency and magnitude. This carrier voltage provides the input for the conversion element in the tracer. This conversion element is essentially a transformer with variable coupling between primary and secondary windings. It serves as a modulator of the carrier voltage. The tracer point is attached directly to the secondary coil and, as it moves in and out of the tracer housing, the coupling between the primary and secondary coils is changed.

The output of the converter is thus a modulated carrier voltage, and the quantity in which we are interested is the amount of the modulation. When measuring irregularities of the order of microinches, this quantity is very small. Therefore the modulated carrier signal is amplified, Fig. 2, to give the desired sensitivity.

After proper amplification, the modulated carrier must go through a suitable demodulating circuit. This serves to eliminate the carrier and leaves only the "measuring signal," which was introduced by the conversion element in the tracer, and which has now been sufficiently amplified for purposes of meter operation.

The measuring signal coming out of the demodulator consists of a varying direct-current voltage, which in turn operates the meter. When the tracer point is in the center of its range, this voltage is sufficient to put the meter pen at the center of the chart. If the tracer point now moves up or down, the coupling between primary and secondary coils in the tracer is changed so that more or less voltage appears at the output of the demodulator. This causes the pen to move correspondingly from its center position on the chart. The amount and direction of this pen displacement is in direct proportion to the amount and direction of the tracer-point displacement.

As already explained, this type of displacement-measuring circuit has no low-frequency cutoff and therefore is not dependent upon mechanical oscillations of the tracer point to give signal voltages. This makes it possible to register on the profile record any of the irregularities which the tracer point encounters as it moves across the surface of the specimen, including waves, bows, steps, or random irregularities. There is a high-frequency cutoff for the meter, but, by selecting an appropriately slow tracing speed, the rate of signal variations may be kept well below the limitations of the meter. Accordingly, all of the roughness and other closely spaced irregularities are shown in true relation to

the waviness and other widely spaced or nonrecurrent irregularities.

The basic control for the measuring circuit just described is the range switch on the Amplicorder control panel, Fig. 3. This is essentially an attenuator which permits the user to select the proper scale of vertical magnification for the surface being profiled. A "zero adjust" control permits fine electrical compensation for a suitable tolerance in vertical positioning of tracer. Table 1 shows the scales in the vertical magnification range. Any of these scales may be selected by means of the range switch shown in Fig. 3.

TABLE 1 SCALES IN VERTICAL MAGNIFICATION RANGE

Amount of vertical magnification.....	×500	×1000	×2500	×5000	×10000
Vertical measurement represented by full chart width, in.....	0.010	0.005	0.002	0.001	0.0005
Vertical measurement represented by smallest division of chart, microinches.....	200	100	40	20	10

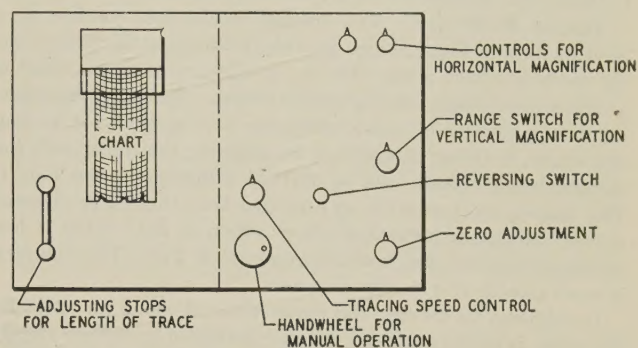


FIG. 3 BASIC CONTROLS ON AMPLICORDER PANEL

Horizontal Magnification. Whereas appropriate vertical magnification in a profile record is obtained by an electronic measuring circuit, horizontal magnification is the result of a mechanical ratio between the tracing speed and the chart speed. A range of horizontal magnification is indispensable in profile recording when various types of surfaces are to be measured. In general, the more widely spaced irregularities, such as waves or bows, require a small horizontal magnification. Otherwise, the chart length for a 2-in. trace becomes inconveniently long. If a chart is too long, it becomes difficult to observe the relationship between the various irregularities. For more closely spaced irregularities, and especially fine roughness, it is necessary to use a much larger horizontal magnification. Otherwise, the many fine irregularities crowd each other on the chart so that they lose their identity.

The ratio between tracing speed and chart speed gives horizontal magnification, and is determined by suitable gearing. To change horizontal magnification, it is necessary to change only the gear ratio. This is done by means of the selective gearbox shown in Fig. 4. Provision is made for eight different ratios between input shaft and output shaft. A pair of knobs on the Amplicorder control panel, Fig. 3, provides for convenient selection of any one of the eight ratios. The settings of these knobs are calibrated directly in horizontal magnification.

TABLE 2 HORIZONTAL MAGNIFICATION RANGE

Ratio of chart speed to tracing speed	
1	20
2.5	50
5	100
10	200

Table 2 shows the values provided. The values in the first column are normally used when studying waviness, while those

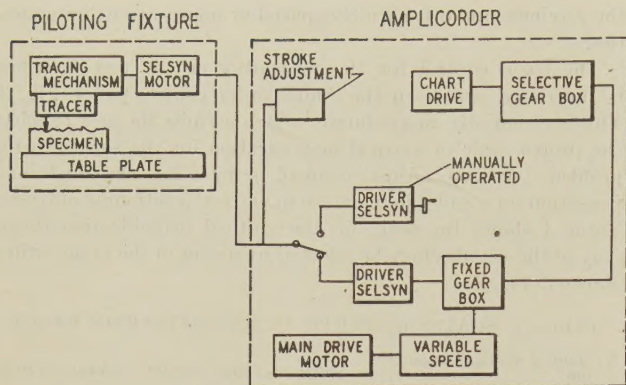


FIG. 4 FUNCTIONAL DIAGRAM SHOWING MECHANICAL ELEMENTS OF PROFICORDER

in the second column are useful for recording detail of roughness.

Tracing Mechanism. The tracing mechanism consists of a piloting fixture, Fig. 4, which provides for moving the tracer over the surface in such a way that the profile drawn on the chart is a true representation of the profile traced. Basically, the piloting fixture consists of a slide-and-screw arrangement for moving the tracer, a tilting mechanism for aligning the tracer with the surface being traced, and a vertical adjustment, see Fig. 1. The tracing mechanism is so arranged that the tracer element moves through a horizontal plane which is flat within a few microinches over a distance of approximately 2 in. The accuracy is much greater over shorter distances.

In addition to moving the tracer element along an optically flat plane, it is also necessary to have the line of trace very closely parallel to that portion of the surface being traced. This is necessary in order to keep the profile record on the chart. Lack of parallelism between the line of trace and the specimen causes the whole profile to move either to one side of the chart or the other. This alignment of the fixture, or tracer, with respect to the surface, must be done for each new specimen. Two adjusting wheels, one coarse and one fine, are provided on the fixture for this purpose, see Fig. 1.

In placing the specimen under the tracer unit, it is necessary to have some means of raising and lowering the tracer. This is accomplished by mounting the entire piloting mechanism on a vertical slide, see Fig. 1. Thus the tracer can be raised high enough to allow for convenient mounting of the specimen to be measured. After the specimen is properly placed, the tracer is lowered by means of the vertical slide until the tracer point touches the surface. In practice, it becomes necessary to make a combination of vertical and aligning adjustments for the final setup. Skill in making these adjustments is readily developed.

The tracing mechanism is driven by a selsyn motor. This motor is driven by one of two driver selsyns in the Amplicorder cabinet. One of the driver selsyns is operated manually by the handwheel for manual operation (see the control panel, Fig. 3). By using this means of driving the piloting-fixture selsyn, it is possible to traverse quickly the length of trace, or to exercise care in moving the tracer point up to a shoulder, or in a slot. It is used primarily for setup purposes. The other driver selsyn is motor-driven, and provides for uniform motion at calibrated speeds.

A range of tracing speeds is desirable for convenient profile recording. If the tracing speed is too fast for a given surface, the meter will fail to follow faithfully the mechanical displacements of the tracer point as it travels over the irregularities of the surface. This results in distortion of the profile drawn by the pen. However, using a tracing speed which is much slower

than necessary for the surface in question results in undue length of time for drawing the record. The variable-speed drive shown in Fig. 4 therefore provides an appropriate range of tracing speed. It is adjusted by means of a calibrated knob on the Amplicorder control panel, Fig. 3. This adjustment provides a continuously variable tracing speed from 0.0005 to 0.015 ips. Since close adjustment is not required, the knob has only six calibration marks within this range. Selection of the proper tracing speed for making a profile record of a given surface depends upon the character of the surface and the type of information desired.

In general, higher tracing speeds are used for waves and other widely spaced irregularities. Fine roughness, however, requires a slow tracing speed so that the high-frequency cutoff of the meter is not exceeded. Experimental evidence has indicated that 0.0005 ips is a sufficiently slow tracing speed to record faithfully the finest roughness which is currently produced by the usual grinding and machining operations.

The maximum length of trace on the Type RLA piloting fixture, Fig. 1, is $17\frac{1}{8}$ in., but in many instances there is no need to utilize the full trace. By means of a stroke adjustment (see Amplicorder unit in Fig. 4), it is possible to select whatever length of trace is desired. Adjusting stops are located on the control panel, Fig. 3. At the end of the selected trace, the power to the piloting-fixture selsyn is cut off automatically, thus stopping the tracer. Provision is also made for a reversing switch, Fig. 3, which makes it possible to drive the tracer in the opposite direction.

The Type RLA piloting fixture, Fig. 1, provides a table plate for mounting the specimen under the tracer. This plate is an integral part of the piloting fixture. Other piloting fixtures are designed so that the piloting mechanism rests directly on the surface itself. The important feature is that there must be no relative vertical movement between surface and piloting mechanism while the record is being taken. The measuring circuit has no means of distinguishing between irregularities of the surface of the specimen and extraneous movements of the specimen. Accordingly, sufficient rigidity must be provided in the mounting mechanism, and in the tracer supporting mechanism, so that extraneous motions are kept within acceptable limits. The following summarizes the functional details discussed in this section:

The Proficorder is an instrument which provides a means for moving a tracer point over a surface, the end result of which is a profile record on a chart. This profile record is characterized by appropriate vertical and horizontal magnifications, which are obtained by electronic and mechanical means, respectively. Physically, the instrument consists of three separate units, i.e., tracer, piloting fixture, and Amplicorder. For the sake of adaptability, the Amplicorder has been designed and constructed as the basic unit which can be used with different types of tracers and piloting fixtures. A line of tracers and piloting fixtures provides the versatility to cover a wide variety of work.

TESTS FOR DETERMINING ACCURACY AND SENSITIVITY OF PROFICORDER

When viewing a profile record showing the irregularities of a given surface, one should know what accuracy can be attributed to this record, i.e., to what extent may the profile record include errors which are due either to inherent limitations or to malfunctioning of the instrument.

There are several basic tests, as follows, which are used in establishing the accuracy and proper functioning of a profile recorder:

- 1 Accuracy of the piloting mechanism is best determined by the profile record obtained from tracing across an optical flat for

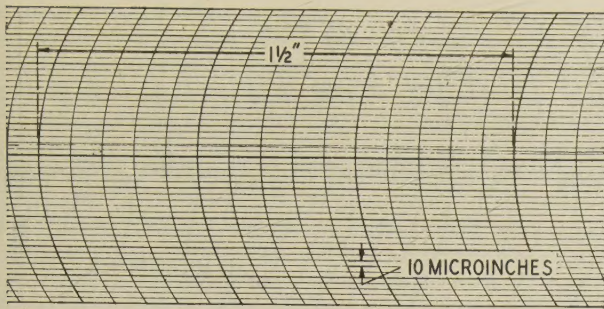


FIG. 5 PROFILE RECORD OF OPTICAL FLAT

(Magnifications: V-10,000, H-5, before 3 to 1 reduction. Basic test for determining accuracy of piloting mechanism. Accuracy for $1\frac{1}{2}$ in. of trace is better than 5 microinches, as determined by this record.)

the maximum trace of the piloting fixture. Fig. 5 illustrates such a record run on the Type RLA fixture. If the optical flat has an accuracy of 1 microinch, any deviations greater than this on the profile record of the flat can be attributed to inaccuracy of the instrument, or to distortion of the optical flat by improper mounting. As determined by this method, the accuracy is approximately ± 3 microinches over a $1\frac{1}{8}$ -in. trace. This degree of accuracy is highly desirable when determining the vertical magnitude of waves, bows, or other widely spaced irregularities. It is especially important when one wishes to determine the relative merits of different machining or finishing processes which are intended to eliminate or cut down the magnitude of certain surface irregularities.

2 Sensitivity of the instrument is most conveniently checked by tracing over a known gage-block step. Fig. 6 shows the profile obtained by tracing over three gage blocks wrung onto an optical flat, side by side. The gage blocks selected provide for steps of 0.0008 in. and 0.0004 in. At V-5000 magnification, full chart width represents 0.001 in. Tracing across the two steps therefore should give a profile showing steps equal to 80 per cent and 40 per cent, respectively, of the chart width. At lower settings of the vertical range switch, the magnitude of the steps on the chart record will be proportionately smaller. Accuracy of the rated magnifications is held within a few per cent.

3 The question also arises as to how faithfully the instrument can repeat itself when going over the same trace a second time. This can be checked by drawing a profile record of a surface which shows a number of distinct irregularities easily identified on the profile. The tracer and chart are then returned to their original starting point. By means of the zero-adjustment control, the pen is moved over on the chart so that a duplicate profile can be

drawn below or above the first record. A second trace is then made over the same surface profile, see Fig. 7. Comparisons between individual irregularities as to size and shape can then be made very readily. The agreement is excellent.

4 Testing for broken or dull tracing points is accomplished as follows: The tracing point commonly used in the tracer is a pyramidal diamond which has a tip radius no larger than 100 microinches. Due to the comparatively large width-to-height ratio of most surface irregularities, this tip radius is sufficiently small to bottom the contour of virtually all the irregularities encountered by the tracer point. Small tip radius is unimportant when measuring waviness because of the large horizontal spacing. For more finely spaced irregularities, experiment has shown that diamonds with a tip radius smaller than 100 microinches produce a profile record which shows scarcely any appreciable differences from that drawn with the 100-microinch tip radius.

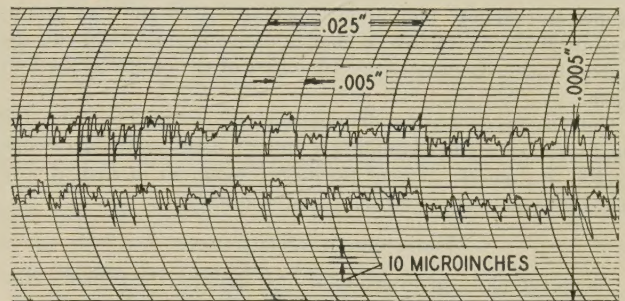


FIG. 7 DUPLICATE PROFILE TRACES

(Magnifications: V-10,000, H-100, before 3 to 1 reduction. This is convenient test for determining whether or not instrument repeats itself faithfully. Individual irregularities are readily compared on the two profiles.)

However, it is possible for these diamond tips to become dull or damaged. This will ordinarily reduce the detail of the closely spaced irregularities, or fine roughness, shown on the chart. The reason for this is that the dull tracing point fails to bottom the fine irregularities on the surface being traced. The profile of ground glass is almost always characterized by a great deal of fine detail superimposed on the coarser roughness. Fig. 8(a) shows a profile of a standard ground-glass specimen. Therefore, if it is suspected that a tracer point is dull or broken, a profile should be drawn of a known ground-glass specimen. If fine detail shows up on the record, the tracer point is probably in good condition. However, if the fine roughness which characterizes ground glass is not present on the profile, Fig. 8(b), the diamond is either worn or broken. In making such comparisons, it is not

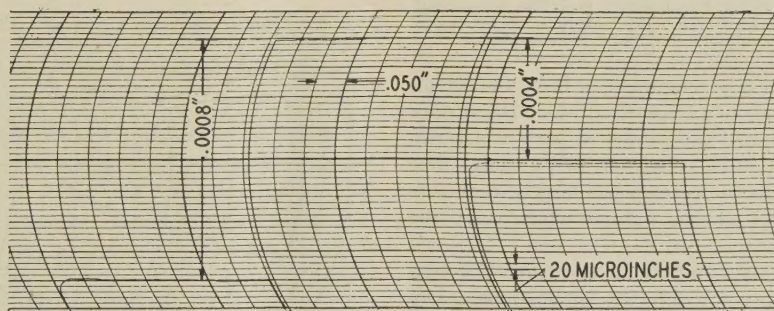


FIG. 6 PROFILE OF GAGE-BLOCK STEPS

(Magnifications: V-5000, H-10, before 3 to 1 reduction. Gage blocks of different thickness wrung onto an optical flat side by side provide steps of known height. Middle block is 0.0008 in. thicker than left block, and 0.0004 in. thicker than right. Tracing over blocks produces profile shown, which indicates sensitivity of instrument. Crack between blocks is due to chamfer on blocks. Deviations of record from theoretical shape include tolerances in gage blocks, uncertainties of mounting of blocks, and tolerances in instrument.)

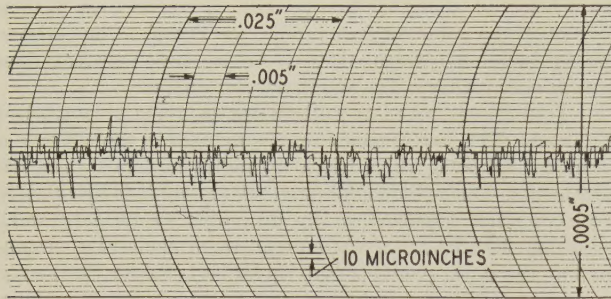


FIG. 8(a) PROFILE SHOWING USE OF SHARP DIAMOND AS TRACER POINT; TIP RADIUS UNDER 100 MICROINCHES

(Magnifications: V-10,000, H-100, before 3 to 1 reduction. Represents characteristic profile of ground-glass surface. Fine detail is lost when tracer point becomes dulled, see Fig. 8b.)

proper to compare individual irregularities on the two graphs because it is not feasible to change diamonds and then run the second trace over exactly the same part of the profile as the first trace.

Conclusions concerning diamonds may be verified by removing the tracer point and examining it under a high-power microscope ($\times 400$ or higher). Tracer points are mounted in such a way that they can be removed and replaced readily.

PROFICORDER APPLICATIONS

As indicated in the introductory paragraphs, the primary consideration in the development of the Proficorder was that of measuring surface waviness. The nature of the instrument is such, however, that it will draw a complete profile of the surface, including all irregularities traversed by the tracer point. This profile record shows the fine surface roughness superimposed upon the waves and other widely spaced irregularities, Fig. 9.

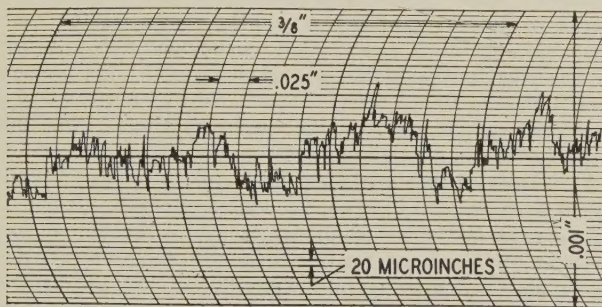


FIG. 9 PROFILE SHOWING ROUGHNESS SUPERIMPOSED ON WAVINESS (Magnifications: V-5000, H-20, before 3 to 1 reduction. This profile represents all irregularities traversed by tracer point. More detail for closely spaced irregularities can be obtained by greater magnification.)

Whether measuring waviness or roughness, or both, the usefulness of the instrument depends to a large extent upon the variety of specimens or actual machined and processed parts which can be profiled in their natural state, condition, or position. The scope of the instrument in this respect depends largely on the nature and adaptability of the piloting fixtures which carry the tracing elements.

As currently developed, therefore, there are three different basic types of piloting fixtures for different applications. Any one of these can be used with the standard Amplicorder unit which contains the amplifier-recorder, the variable-speed drive mechanism, and all necessary controls for operating the piloting fixtures.

The Type RLA piloting fixture, Fig. 1, has been described at some length in this paper. It is mounted on a base plate which

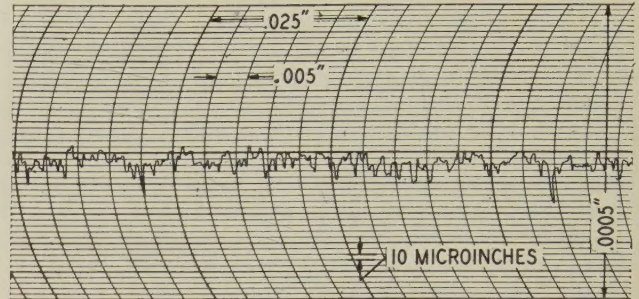


FIG. 8(b) PROFILE SHOWING USE OF DULL DIAMOND AS TRACER POINT; TIP RADIUS OF 600 MICROINCHES

(Magnifications: V-10,000, H-100, before 3 to 1 reduction. Trace was made over same ground-glass specimen, not identical profile, as used for Fig. 8a. Note loss of fine detail as compared with profile in Fig. 8a.)

rests on legs. The fixture is stationary in character. It can accommodate a great variety of parts which are of such size that they can be brought to, and placed on, the table plate under

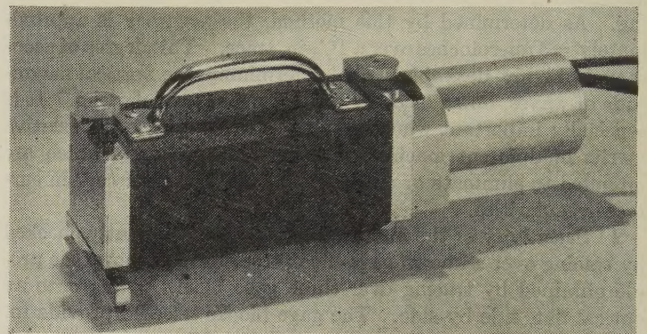


FIG. 10 TYPE RPA PILOTING FIXTURE

(Used primarily for large outside diameters and for measuring flats which are not readily moved from their location. Fixture is mounted directly on specimen. Over-all dimensions $3\frac{1}{4}$ in. wide \times $4\frac{1}{2}$ in. high \times 12 in. long.)

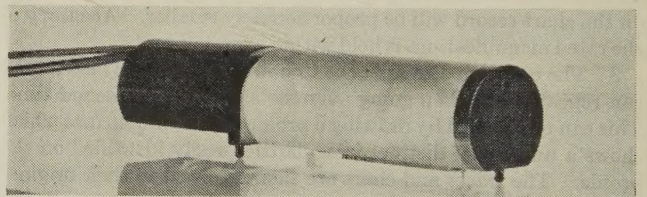


FIG. 11 RCA PILOTING FIXTURE FOR INTERNAL DIAMETERS (Used for taking profiles of cylinders.)

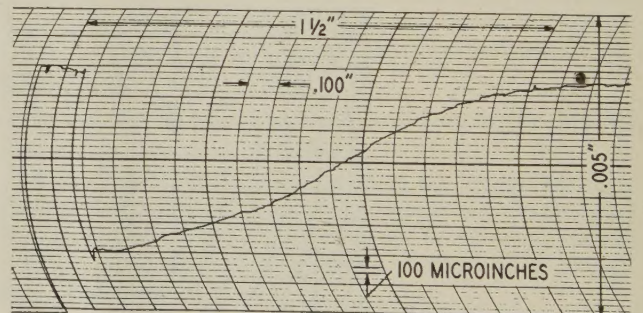


FIG. 12 PROFILE RECORD OF WORN CYLINDER WALL IN USED AUTOMOBILE BLOCK

(Left end of record represents top of cylinder. Magnifications: V-1000, H-5, before 3 to 1 reduction. Step shows where piston ring stopped when piston reached top center position. Higher compression, poor lubrication, starting friction, all account for greater wear as piston approaches top of cylinder.)

the tracer unit. Profile records may be drawn of outside diameters (axially) as well as flats. For large outside diameters, such as rolling-mill rolls and flats which cannot be moved conveniently, or are too large to be placed on the table plate of the Type RLA piloting fixture, there is a portable piloting fixture, Fig. 10. This fixture can be carried to the specimen and placed directly on it. Sufficiently long cables for connection to the Amplicorder unit are attached to the fixture. The necessary controls for leveling and vertical adjustment are on the fixture.

A third fixture provides for measuring profiles in an axial direction on inside diameters such as cylinder walls, Fig. 11. The fixture is placed directly inside the cylinder and can be used for $2\frac{7}{8}$ in. ID or larger. This fixture has a stroke of $2\frac{1}{2}$ in., which is somewhat longer than that of the other two fixtures. Fig. 12 shows the profile taken axially along the cylinder wall of a used automobile block. The step shown on the record is where the upper ring stopped during each stroke of the piston.

MEASURING SMALL ANGLES RELATIVE TO A GIVEN PLANE

This is a problem somewhat different from the one usually associated with the drawing of surface-profile records. However, the angle which one part of a surface makes with another part is a matter of profile, and as such, the Proficorder provides a means for measuring this angle. When the angle is very small, the Proficorder can be used to obtain a high degree of accuracy in its measurement.

The profile record in Fig. 13 serves as an illustration. The setup is made in such a way that when tracing across the so-called reference plane, the line or profile drawn on the chart lies parallel to the lines of the chart and very near to one edge. As the tracer point moves over the plane which makes an angle with the reference plane, the line drawn on the chart will slope toward the other edge of the chart. When the pen reaches the

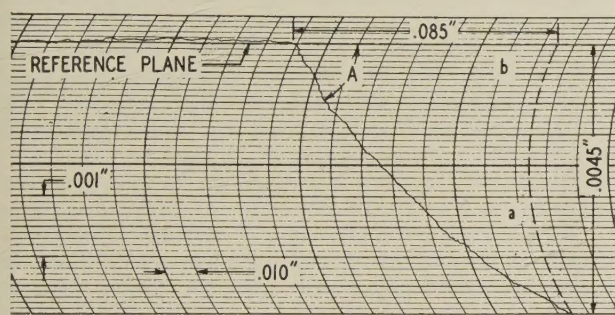


FIG. 13 PROFILE RECORD ILLUSTRATING USE OF PROFICORDER AS ANGLE-MEASURING DEVICE

(Magnification: V-1000, H-50, before 3 to 1 reduction. Represents trace over two plane surfaces which meet at an angle. Angle A between the two is a function of a and b ; a and b are scaled from chart.)

opposite side of the chart, the tracing is stopped. The chart now has on it a line with a certain slope; the amount of slope being dependent upon the size of the angle and the horizontal and vertical magnifications used. This line may be considered as the hypotenuse of a right triangle. From the illustration, Fig. 13, it is evident that the angle A , in question, is a function of the sides a and b . The sizes of a and b are scaled from the chart. Therefore the value of A is easily obtained from the following trigonometric relation

$$\tan A = a/b = \frac{0.0045}{0.085} = 0.0529$$

$$\text{Angle } A = 3^{\circ} 2'$$

SUMMARY

The Proficorder represents a major addition to industrially available surface-measuring instruments. It provides data which are highly essential in the study of surfaces and the processes by which they are obtained. It is unique among surface-measuring instruments because of the following characteristics: (1) It reproduces, with appropriate magnifications, the actual profile of a considerable length of surface. This gives a profile record in which finely spaced irregularities, such as roughness, are superimposed on the more widely spaced irregularities, such as waves or bows. (2) It permits the study of individual irregularities as to size, shape, and relative position with respect to other irregularities.

The implication of these two features is that the fields of application are very wide. In the industrial process, there are three general areas of usage which suggest themselves, as follows:

- 1 The instrument can be used to good advantage for inspection of production setups for waves and other surface characteristics. These surface characteristics are determined, not only by the general condition of the machine (bearings, spindles, gears, balance, etc.), but more directly by the nature and condition of the cutting edges of tools and the cutting grits of wheels. It is next to impossible to specify these cutting edges; moreover, they may change significantly from setup to setup and during ordinary runs. Use of the Proficorder during machine setups allows these variables to be controlled, thus providing more consistent quality and preventing loss of time and material.
- 2 As an instrument for use in process engineering, it provides quantitative data for determining the effectiveness of steps taken to improve or obtain certain surface finishes in machining or grinding operations. Moreover, it can be used to good advantage in determining the effect of subsequent finishing processes on surface character, particularly waviness and other widely spaced irregularities.
- 3 As a laboratory instrument, the Proficorder supplies indispensable information to all those who are interested in determining the causes, effects, and characteristics of surface irregularities.

Discussion

H. BLOK.³ In recording a surface profile, the conventional methods, including that employed in the Proficorder, consist of plotting the ordinates (heights) of the irregularities as a function of the abscissas; for further reference it is proposed to call this the Cartesian method.

In a common variant of the Cartesian method, the scale of magnification for the ordinates is chosen larger than for the abscissas, because the mean slope of the irregularities as a rule is small and otherwise a great length of recording paper has to be used. The gain in surveyability outweighs the disadvantage of the distortion of the record.

It is felt that further improvements may be achieved by changing over to another method of recording, i.e., to the polar-diagram method. The polar method is well known in the field of wave analysis (for example, analysis of electric currents), but apparently has not yet received the attention from workers in the field of surface roughness which it undoubtedly deserves.

In contrast to a Cartesian diagram, in a polar diagram the ordinates y are not plotted as a function of the abscissas x , but as a function of the slopes dy/dx of the irregularities. For example, for a sinusoidal surface profile, the polar diagram proves to be an ellipse; by proper adjustment of the scales, the ellipse may be transformed into a circle. Going a step further,

³ Royal Dutch Shell, Delft, Holland.

it can easily be shown that for each truly periodic surface profile, which possesses no discontinuities in height or slope of the irregularities, a polar diagram will be found which consists of a single closed curve. In general, it can be proved that, for the purpose at hand, the polar diagram characterizes the surface profile mathematically as completely as the Cartesian diagram. It is true that the polar diagram cannot provide the level of the surface profile relative to some reference level but for this purpose, i.e., studying surface roughness, this is wholly unimportant.⁴

The polar method is not just a mathematical trick, but it presents several distinct advantages over the Cartesian method, particularly if the surface profile is to be studied in terms of statistics; some advantages are as follows:

- 1 The polar diagram occupies much less space than the conventional Cartesian diagram.

- 2 In studying the distribution (statistically speaking, the frequency diagram) of the slopes occurring in the surface profile, the polar diagram provides a more direct approach than the Cartesian diagram. In the writer's opinion, the distribution of

⁴ Mathematically, the polar diagram relates the ordinates y to the corresponding values of dy/dx . From such a relation (an ordinary differential equation of the first order), the relation between y and x , i.e., the surface profile, can be found by integrating once, for example, by well-known graphical methods. In this integrating process one indeterminate constant of integration is introduced: This feature explains why the polar diagram fails to provide the exact level of the surface profile.

slopes is in many cases no less important than the distribution of heights.

- 3 In studying deviations from true periodicity of the irregularities, the polar diagram is more convenient than the Cartesian diagram; in the polar diagram such deviations result in a branched curve intersecting itself and delineating a ring-shaped region. From the width of this ring the magnitude of the deviations may be judged at a glance.

- 4 In combination with item 3, for the purpose of comparing the profile actually obtained in the finishing process with the profile prescribed, the polar diagram is more convenient than the Cartesian diagram. Whereas interpreting polar diagrams in terms of the actual surface profile requires experience and skill, for inspection purposes quite inexperienced personnel may be employed, as only a ring-shaped template with clearly indicated "tolerance widths" has to be compared to the polar diagram obtained.

Finally, it should be remarked that polar diagrams may be plotted automatically by relatively simple electrical means.

AUTHORS' CLOSURE

Mr. Blok's mathematical transformations are interesting. Thus far the authors' purpose has been to develop a simple, rugged, versatile instrument for shop use. The profiles obtained show the actual surface irregularities in true relation to each other, and thus far the interpretations by shop personnel have been simple and straightforward.

Development of an Air-Operated Force-Measuring System

By A. A. MARKSON¹ AND R. S. WILLIAMS,² PITTSBURGH, PA.

The force-measuring system described in this paper depends upon the principle of using a nonmetallic flexible diaphragm as a null-deflection frictionless measuring piston. This is accomplished by use of a simple auxiliary positioning pneumatic-relay system. A novel viscous stabilizing damper is described. The paper outlines the industrial development of this principle. Previous related NACA work in this field is cited. Because the devices are now used in test measurements on cradle dynamometers, turbojet engines, rockets, and guided missiles, some essential data relating to the evaluation of such a system and its probable errors are given. It is shown that this principle is capable of yielding an unusually good combination of ruggedness and stable accuracy.

INTRODUCTION

THE use of flexible nonmetallic diaphragms as measuring elements is not new. Industrial draft gages, for example, employ this principle extensively. The development of a good force-measurement system, using a flexible diaphragm, in which the measurement is obtained as the product of a balancing air pressure acting upon the diaphragm area, involves problems to a degree not ordinarily encountered in usual good diaphragm-operated instruments.

A particular engineering development in this field of measurement, the subject of the present paper, is known commercially as the "ThrusTorq" system of force measurement. Figs. 1 and 2 are schematic diagrams which show the elementary devices and their method of operation. As force is applied to the measuring diaphragm of Fig. 1, a relay pilot valve automatically admits or exhausts air from the diaphragm chamber, acting as a null-point regulator of the diaphragm position. The balancing air pressure is suitably measured and, if the area of the diaphragm is satisfactorily constant, is a measure of the applied force. The system will be recognized as one in which instability is possible in the absence of damping. The required damping is supplied by a viscous damper.

DETAILS OF SYSTEM AND OPERATION

Referring to Fig. 1, the construction of this unit consists of a molded diaphragm the outer edge of which is clamped between the diaphragm cage and cover. Clamped to the diaphragm is a rigid center which carries the exhaust port. The rigid center or thrust plate is attached rigidly to the force arm. A force applied to the force arm is transmitted to the measuring diaphragm by the arm and the elastic fulcrum. The stabilizer capsule, attached to the force arm, reacts with the instrument frame through the stabilizer thrust buttons. Located in the diaphragm cover is the

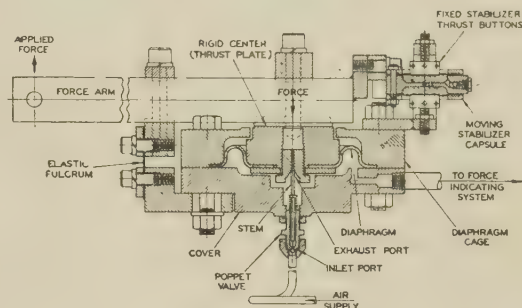


FIG. 1 SCHEMATIC DIAGRAM OF THRUSTORQ SYSTEM OF FORCE MEASUREMENT

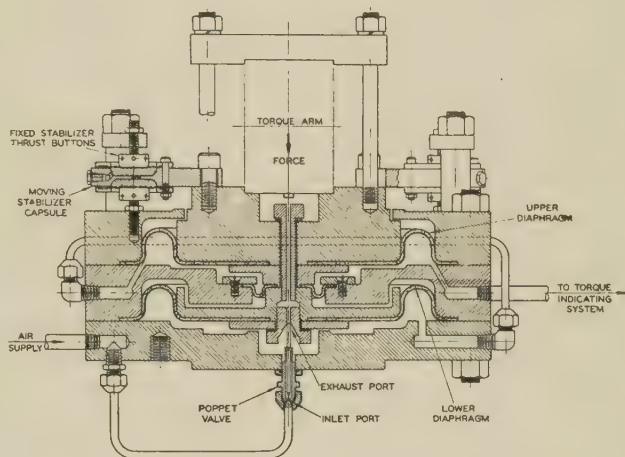


FIG. 2 METHOD OF OBTAINING AREAS IN SERIES

double-seated poppet valve which acts as a null-position regulator of the diaphragm position.

The operation of the system can be described by first explaining the poppet-valve construction and operation. The poppet consists of a solid stem, one end of which is formed into an inlet poppet head, the other end serving as an exhaust-valve stem. When the unit is in operation, the inlet poppet is seated in the inlet port. The exhaust end of the valve seats in an exhaust port which is carried in the rigid center. With compressed air connected to the poppet-valve inlet, a movement of the diaphragm toward the poppet valve under the action of increased load causes the inlet valve to open, the exhaust remaining closed. This admits air to the diaphragm chamber.

The air pressure acting on the diaphragm area causes the diaphragm to return to the null position, closing the inlet. A decrease in load on the force arm allows the air pressure existing on the diaphragm to move it away from the poppet valve, thus opening the exhaust port, while the inlet remains closed. As the air pressure bleeds enough to re-establish the balance between the pressure and the applied load, the diaphragm returns to the null position, closing the exhaust. Thus with an inlet port which is fixed in relation to the instrument frame, the diaphragm will be

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² Production Engineer, Hagan Corporation. Jun. ASME.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 47-A-37.

regulated to a fixed equilibrium position under all conditions. The desired equilibrium position for the diaphragm is obtained by adjusting the position of the inlet port of the valve in relation to the instrument frame.

The balancing air pressure generated by the relay-valve action is a linear measure of the applied load. This pressure is taken in practice to standard types of pressure-indicating systems, remotely located in many cases, on which the final load readings or records are made. The output pressure may serve simultaneously as a pneumatic signal for the operation of regulating and computing systems.

This system will be readily recognized as one in which instability is possible when inertial forces are balanced hydrostatically in the absence of sufficient damping. The viscous stabilizer previously referred to is introduced to control this situation. This unit consists of the capsule in Fig. 1, composed of two diaphragm chambers enclosed by flexible diaphragms and connected by a small orifice in the partition plate. The capsule is completely filled with damping liquid. The capsule is attached rigidly to the arm and moves with the arm. With the diaphragm and force arm set in the null position, the thrust buttons are adjusted symmetrically so that they engage the capsule diaphragms, thereby serving as rigid centers for the capsule diaphragms. With the unit so adjusted, any movement of the force arm will cause liquid to be transferred through the orifice from one chamber to the other, thus providing the necessary viscous damping for complete stability. This unit will be recognized as a type of dashpot from which static friction has been removed and alignment troubles practically eliminated. Stable damping constants are obtainable by the use of damping fluids of the silicone family, owing to complete control of damping by the orifice, since piston leakage is eliminated. In further contrast to a piston dashpot, equally high damping forces are obtained in both directions as there is no barometric pressure limit such as determines the maximum force on the suction stroke of the ordinary dashpot. This permits the application of critical damping with the result that response speed is limited principally by the pressure-gage speed.

Fig. 2 shows a variation of the basic elements. Here, two measuring diaphragms have been connected in series to give higher load capacities per unit. This particular unit is capable of balancing directly an 11,600-lb force. The series construction is accomplished by the use of a third small diaphragm as a flexible seal between the two main diaphragms, thus maintaining a frictionless construction.

Since the applications of this system are now numbered by several hundred, an idea of the present field of application may be obtained from some typical installation views, Figs. 3, 4, and 5.

Fig. 3 shows a basic unit mounted on a horizontal turbojet-engine thrust stand. In the foreground appears one of the four elastically suspended stand supports. This particular support design uses a strip of spring sheet in the form of a loop with the two ends secured to the support pedestal. Carried in the loops are the pin shafts of the frame on which the jet engine is mounted for testing. This engine mounting is quite rigid to lateral motion but very flexible along the thrust axis. This type of support is used very effectively with a null-displacement measuring system, resulting in a completely "frictionless" method of measurement.

Fig. 4 shows a basic unit applied to a cradled electric dynamometer used for automotive-engine testing. In this particular setup, it is desirable to operate the system at an angle to simulate the actual operating angle of the automobile engine. Here the dynamometer is mounted at 10 deg to the horizontal. The force-measuring unit on this dynamometer is a double ThrustTorq, consisting of two complete upper and lower diaphragm units

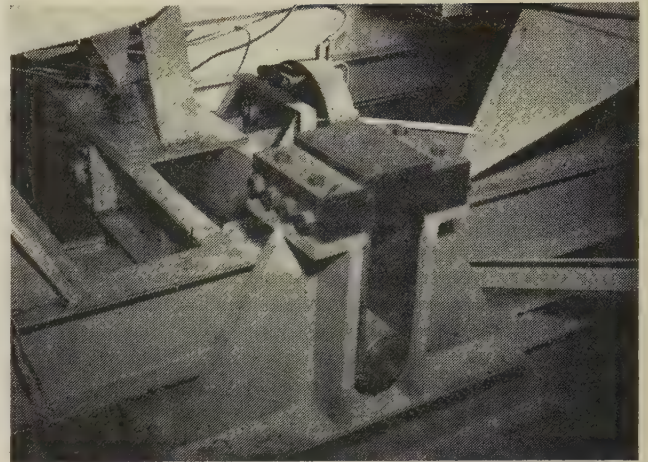


FIG. 3 ELASTICALLY SUPPORTED TEST STAND

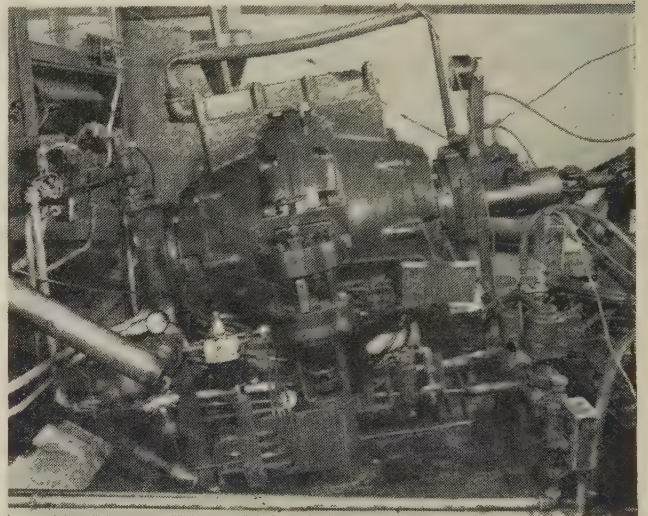


FIG. 4 CRADLED DYNAMOMETER

which allow measurement of forces in either direction. This may also be accomplished using a single unit having a preloaded up-scale zero on the recorder or manometer.

Fig. 5 shows an installation of the integral-elastic-fulcrum device of the type shown in Fig. 1, applied to a batch-weighing hopper installation. A mercury manometer, indicating the weight in pounds, is equipped with an adjustable electric contactor which automatically cuts off the feed to the batch hopper at the desired weight indication.

DEVELOPMENT BACKGROUND

The first serious attempts to produce a satisfactory design of pneumatic-force instrument, which would be an advance in many fields of force measurement over previous devices, were reported by Moore, Biermann, and Voss,³ of the Aircraft Engine Research Laboratory, Cleveland, Ohio.

Six machines of the type shown in Fig. 6,⁴ were built by the authors' company for NACA from NACA designs. These machines used a flat fabric-inserted rubber diaphragm as a meas-

³ "The NACA Balanced-Diaphragm Dynamometer Torque Indicator," by C. S. Moore, A. E. Biermann, and F. Voss, NACA RB No. 4C28, March, 1944.

⁴ Ibid., Fig. 6.

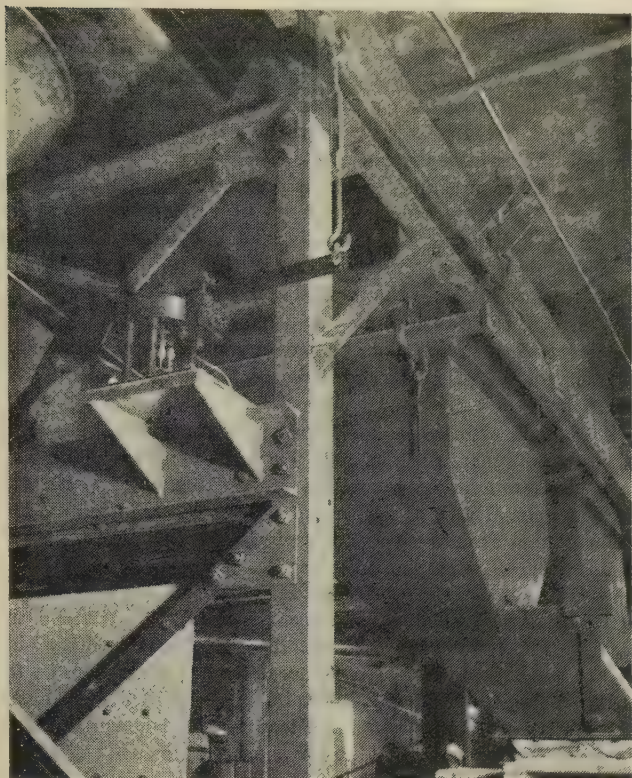


FIG. 5 WEIGHING HOPPER

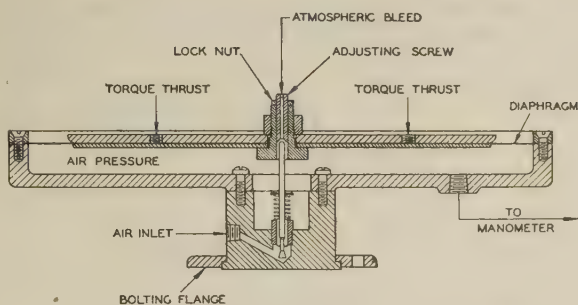


FIG. 6 NACA TORQUE-INDICATOR DESIGN

uring piston of approximately 67 sq in. effective area. In order to minimize the effect of diaphragm displacement on the linearity of the output, a nonbleed relay pilot valve, designed by the authors' company, was incorporated in the design shown in Fig. 6.

As in the authors' development, the valve assembly consists of a valve pin with a poppet head seated in an air-inlet-valve seat. The other end of the pin is seated in an atmospheric-exhaust-port hole which is part of, and can move with, the measuring diaphragm. As the inlet opens, compressed air flows into the diaphragm chamber and closes the valve. Following the instant that the poppet pin is fully seated in the inlet, a further withdrawal of the exhaust seat opens the exhaust port and lowers the pressure till the diaphragm returns, closing the exhaust port. The diaphragm is in positional balance when both exhaust and inlet ports are closed.

In the NACA design, the exhaust port is adjustable in the diaphragm assembly. Since the equilibrium position of the diaphragm is determined by the fixed valve length between exhaust and inlet seats, the equilibrium diaphragm position is adjustable by screwing the exhaust port in or out. When the exhaust port is

screwed toward the inlet, the diaphragm moves in the opposite direction.

The unbalanced valve area is about 0.01 sq in. Errors from this unbalanced area are negligible so that relatively large changes in supply pressure have no effect upon the output of the device.

The valve, being frictionless, produces no measurable effect on the sensitivity which is high. In the steady state, practically no air is consumed.

The advantages of this design are good sensitivity, good linearity, insensitive to minor misalignment.

A typical NACA dynamometer-calibration result is shown in Fig. 7, as taken from the NACA report cited.³

The sensitive performance of the NACA instrument and its convenient manipulative properties bring out the attractiveness of pneumatic systems for force measurement. However, limitations in the performance of this particular instrument design for

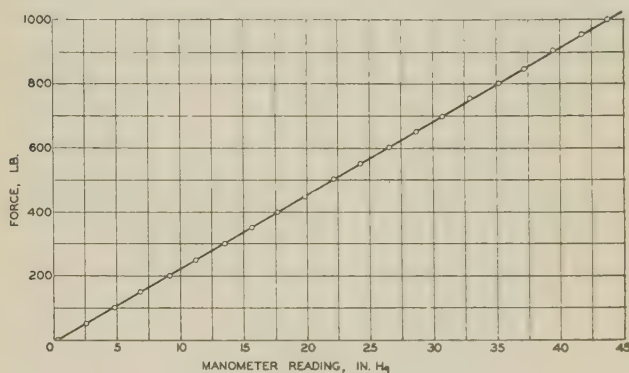


FIG. 7 TYPICAL NACA RESULTS

general application soon became apparent and it was evident that these would have to be overcome before a generally successful industrial system would result.

The limitations other than commercial may be listed as follows:

- 1 Low load-carrying capacity, as compared with other hydrostatic systems; such as, pistons and liquid capsules.
- 2 Instability under inertial loads.

The development of a system overcoming these limitations has been carried out by the authors' company since 1943, and has resulted in the pneumatic force-measurement devices of the present paper.

Since these devices have attained considerable importance in many widely varied fields, ranging from industrial-belt-weightometer applications to research on rocket propellants, a fairly complete design account of this development is justified.

In developing the necessary accuracy data for the design of a measuring system, two general methods of analysis are usually employed. From the designer's point of view, the nature and magnitude of the errors in a well-designed instrument may be considered as being under reliable control within certain limits rather than as being subject to absolutism relating to presence or absence. From this point of view, a knowledge of the performance of the system components is required. Sufficient information of this type suffices, at least in theory, for the synthesis of an instrument. However, the over-all performance of the instrument, as constructed and as actually used is the final criterion for more than platitudinous reasons. Only then does the designer have the final knowledge of how all known and unknown component errors combine in practice to give over-all performance. The subject measuring system will be discussed from both viewpoints.

The potential sources of measurement error in the air-operated measurement system of the paper can be listed as follows:

- 1 Characteristics of the measurement diaphragm with respect to its effective area and stability.
- 2 Influence of the relay pilot valve and the viscous stabilizer on measurement accuracy.
- 3 Performance of the final force-reading devices, such as manometers and pressure instruments; but, because this covers well-trodden ground, it can be omitted in the interest of conciseness.

EFFECTIVE AREA OF NONMETALLIC DIAPHRAGMS

The effective area of a nonmetallic diaphragm used as a force piston may be defined as that area which multiplied by the differential pressure acting on the entire diaphragm gives the true measure of the normal forces resisted by the diaphragm system. To bring this out clearly, consider the diaphragm in Fig. 8. Part of the hydrostatic diaphragm forces are balanced by the load W , and part are absorbed by the diaphragm clamp ring. Eaton and Buckingham⁵ have analyzed this situation. By assuming as they

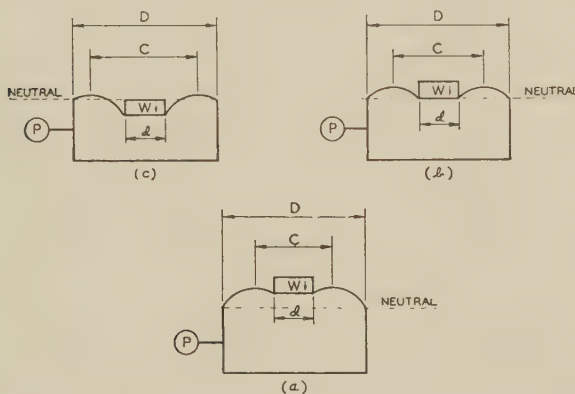


FIG. 8 SHAPE OF SLACK DIAPHRAGMS UNDER VARIOUS LOADS

did that the diaphragm is a limp material incapable of force transmission, except by tension, it is easy to see (a) that all hydrostatic forces acting inside the dimension C must be transmitted to the load and all forces outside C are taken to the rim. It is readily seen that the diaphragm position in Fig. 8(a), has a smaller load-supporting area than that of Fig. 8(b), which in turn is less than that of Fig. 8(c). In other words, the effective area decreases as the weight is raised. Position, Fig. 8(b), is unique, according to Eaton and Buckingham, in that the effective areas of all slack diaphragms which could be installed in the given assembly are generally equal only at this point, which the present authors call the diaphragm neutral. The change of effective area with respect to displacement is a function of the depth of the cup or belly. In the neutral position the effective area is predictable within fairly close limits from the mean diameter $(D + d)/2$.

Therefore the use of a slack diaphragm as a deflecting system connotes a nonlinear system. The degree of nonlinearity is a function of the diaphragm displacement from neutral and of the slackness ratio, other things being the same.

Further, the ideal slack diaphragm of Eaton and Buckingham has a zero mechanical spring rate, that is, it is completely limp.

The optimum conditions for reproducible use of a flexible diaphragm for linear-force measurements are readily perceived. (1) The diaphragm should be used in its neutral position and not

permitted to deflect. (2) Any diaphragm-spring forces should be a minimum in this position. (3) The slackness ratio should be high and not depend upon the manner of installation so that small displacements from the neutral will result in small and controllable errors. (4) Inelastic hysteresis effects can be eliminated by not permitting large displacements.

The diaphragm employed in the present development consists of a molded fabric which is covered by rubber bonded to the fabric. The diaphragm is relatively thick as compared with those discussed by Eaton and Buckingham, the assembly being approximately $1/16$ in. thick in the molded belly.

However, such an assembly is capable of operating satisfactorily at ratings up to 100 psi air pressure, which represents a load capacity of 100 psi of effective area or 11,600 lb for the unit in Fig. 2.

The diaphragm has a finite spring rate and a definite area-displacement characteristic. Fig. 9 is a plot of experimental data on a diaphragm having a nominal area of 3.5 sq in., and a reproducible neutral area as molded of 3.53 sq in. Plotted against displacement from neutral are spring force and effective area determined at constant displacement. The error introduced by the change in the equivalent area with displacement is estimated quite readily. The effective-area curve is quite linear about the neutral point, and for 0.050-in. displacement on either side shows that the effective area changes at the rate of 0.1 sq in. for 0.100-in. displacement, or 0.001 sq in. per 0.001-in. displacement. If the actual displacement under load is kept below, say, 0.005 in. by the relay system, the limit of change of area will be 0.005 sq in., or under 1.5 parts per thousand. Since the displacement may be considered the result of over-all positioning and deflection characteristics of the system, the result is a slight decrease of effective area, and the resultant measurement will be systematically high with respect to the neutral characteristic.

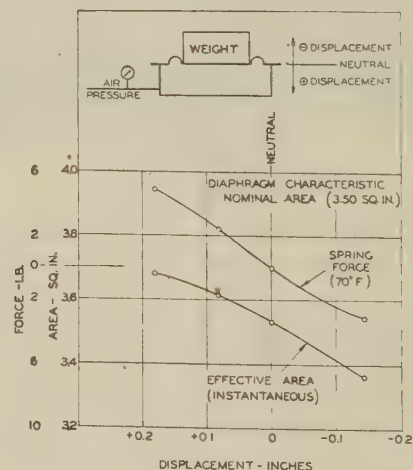


FIG. 9 TYPICAL CHARACTERISTICS OF AUTHORS' DIAPHRAGMS

The diaphragm-spring force over the deflection interval around zero is 1.3 lb per in., or 0.013 lb per 0.001 in. For a 0.005-in. displacement, the diaphragm-spring force would be nearly 1 oz. For the case cited, this error happens to oppose the area-change error. However, the spring force of a rubber diaphragm is inconstant with temperature. Data on the variation of elastic properties of rubber compounds has been given by J. D. Morron.⁶ Fig.

⁵ "Non-Metallic Diaphragms for Instruments," by H. N. Eaton and C. T. Buckingham, NACA Report No. 206, 1924.

⁶ "The Evaluation of Rubber and Rubber-Like Materials," by J. D. Morron, presented at the Semi-Annual Meeting, Detroit, Mich., June 16-19, 1946, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

10⁷ shows typical curves for Buna N which indicate that temperature changes down to 32 F will produce spring errors due to the stiffening of the rubber. However, these errors are quite negligible in their effect on the measurement. For changes extending much below zero F, the resultant errors are more serious and can produce instrument drifts of as high as 1 part in 500 from zero F to -30 F.

When the measuring system is attached to an elastically supported measuring stand such as in Fig. 3, the design of the particular elastic system to give a negligible error for shifts in the mechanism producing an extraneous deflection of 0.003 in. was sufficient to insure satisfactory stability of calibration. Thus assume, in the 3.5-sq in. diaphragm system under discussion,

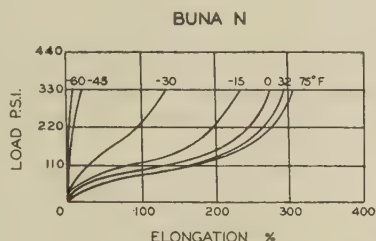


FIG. 10 TYPICAL TEMPERATURE DATA ON BUNA N

that the full-load output is 350 lb force. If the suspension is designed with, say, a force rate of 50 lb per in. in the direction of thrust, the force variation for 0.005 in. would be 4 oz. If the extraneous displacement of the system is a pure and stable load function, the effect on linearity is zero, the spring rate simply becoming a small constant of the system. However, if the displacements are random, as might be caused by shifts in the entire measuring structure, handling these effects, which in the case cited could be as large as 1 part in 1400 of the full load, they may require careful attention when striving for the limit in accuracy.

⁷ Ibid., Fig. 27.

As the instrument is essentially linear, such effects will show up directly as small zero shifts.

PILOT-VALVE INFLUENCE

The pilot-valve area may be taken as 0.01 sq in. The unbalance of the valve is a constant of the instrument but is subject to small variations associated with the absolute tightness of the valve. It has been ascertained experimentally that the parasitic valve effects may be kept satisfactorily low by limiting the minimum size of the measuring diaphragm to 3.5 sq in. The critical over-all test of pilot-valve effect consists of subjecting the instrument to a 10 per cent change in air-supply pressure. Resulting output change should not be readable.

LABORATORY TESTS OF EFFECTIVE AREA

Since the constraints operating on the diaphragm system itself are free of coulomb friction and lost motion, the sensitivity at all load points should be independent of the actual load. This turns out to be the case. For example, using a water manometer as the indicator, a repetitive deflection at any load point is readily obtained by applying a load of 1 part in 50,000 of the full load rating. This figure was determined by the technique described more fully in the discussion of Figs. 11 and 12 which follows. This means that the limit of sensitivity is determined by the sensitivity of the external dynamometer system and the pressure-indicating system. While extreme sensitivity should not be confused with stable accuracy, high repetitive sensitivity is a valuable instrument characteristic in force-measurement systems because the frictional performance of the load-applying system can be most readily evaluated by use of a frictionless measuring system.

A considerable number of field investigations have been made by users of this system which are generally confirmatory of the characteristics discussed.

Fig. 11 gives the results of laboratory studies made by the authors based on the following test method: If a hydrostatically loaded diaphragm of area equal to the measurement diaphragm is

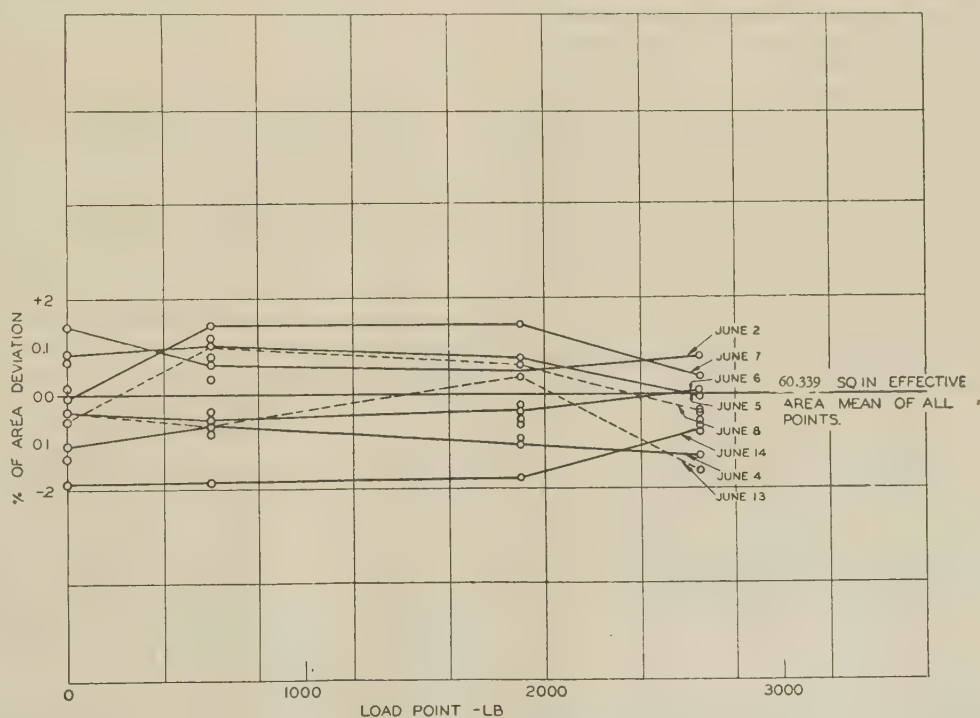


FIG. 1 LONG-TIME ACCURACY AND LINEARITY CHARACTERISTICS OF 60-SQ IN. NOMINAL AREA INSTRUMENT

used to apply full load to the measurement diaphragm, then a water manometer may be connected differentially across the two diaphragms and will assume a zero reading corresponding to the weight of the parts carried by the measurement diaphragm. The smallest weight added to the measurement diaphragm, which will give an additional deflection on the manometer, yields the sensitivity at the time when the measurement diaphragm is under its full rated load output.

Fig. 12 shows the system schematically. A double unit is used, the lower unit being the measurement unit under test. The upper unit pilot valve is removed and the exhaust blanked. Air pressure is applied to the upper unit by means of a precision dead-end reducing valve, and the hydrostatic force is transmitted

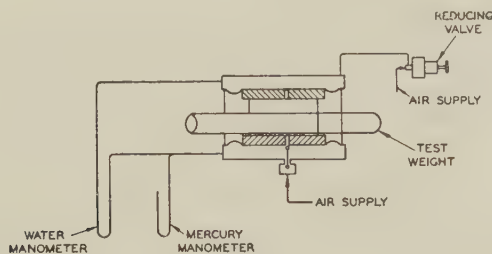


FIG. 12 LABORATORY TEST METHOD FOR DETERMINING EFFECTIVE AREA

to the lower unit which generates an output pressure, observed on a mercury manometer. A 100-in. water manometer is connected differentially between the upper and lower diaphragms. A test bar, carefully compared with class B weights and of a weight sufficient to give an output change of 70 in. of water on the measurement diaphragm, is then poised on the system. The application of the test weight causes the output of the system to increase, and the increase is read on the differential water manometer. The instantaneous effective area can be calculated from these data and plotted against the observed mercury-manometer output, which gives the load point at which the effective area was determined.

An investigation of the 60-sq in. unit was made, using the foregoing procedure over a 2-week period with the results shown in Fig. 11. This figure shows the mean-effective-area curve for the diaphragm, as determined from the individual daily runs which are also given. Between runs, maximum loads indicated were left on the instrument. No instrument adjustments were made during the test period, the principal object being to determine the reproducibility of effective area over a relatively long time. Field tests generally confirm the character of these results.

These results present a factual representation of the probable long-time stability of effective area and linearity characteristic, as determined by observations of a care and character found in industrial laboratories.

OPERATING EXPERIENCE

As mentioned, several hundred of these units have been placed in service. These installations have been used principally as test indicators and recorders on dynamometer systems of various kinds ranging from electric-cradle units to buzz-bomb and rocket-engine stands. Industrial applications have been very few and principally limited to automatic weigh-bin and conveyer operation.

The principal difficulties encountered in the applications of these units have been as follows:

1 Mechanical troubles with the stabilizer units gave the most concern before complete liquid tightness and freedom from dia-

phragm breakage were finally achieved. It was found that in some cases the force arm lacked sufficient torsional rigidity, necessitating a design change on one frame size so that the stabilizing force was applied in line with the applied force.

2 It was found that occasionally a unit would be attached to a poorly designed testing mechanism, with the result that friction and lack of repetition would be attributed to the novel feature of the setup. An occasional unit would be badly installed or poorly adjusted.

3 The probity of an instrument reading force on a mercury column or other pressure instrument had to be established to the satisfaction of the user of established methods.

These experiences have not been one-sided by any means. The advantages of a simple frictionless diaphragm device using air and capable of the remote transmission, control, and recording of force measurements with acceptable accuracy have been sufficient to win a degree of acceptance by high-grade laboratories, which encourages further perseverance in this development.

Discussion

N. B. NICHOLS⁸ AND E. H. WOODHULL.⁹ It is believed that a more detailed examination of the pilot-valve reactions is desirable inasmuch as any nonbleed-type pilot yet encountered by the writers has an inherent dead zone about the equilibrium position. Neglecting the secondary forces produced by Venturi effects caused by flow through the ports, the following force equations can be written for the pilot valve shown in Fig. 6 of the paper; with the exhaust port closed and the inlet port opened an infinitesimal distance

$$F_1 = P_B(A_D - A_E) + P_s A_I + F_0 - P_B[A_I - A_E] \dots [1]$$

With the inlet port closed and the exhaust port opened an infinitesimal distance

$$F_2 = P_B(A_D - A_E) \dots [2]$$

$$\Delta F = F_1 - F_2 = P_s A_I + F_0 - P_B[A_I - A_E] \dots [3]$$

where

F_1, F_2 = force applied to diaphragm by force arm

A_D = effective area of diaphragm

A_I = area of inlet port

A_E = area of exhaust port

F_0 = initial tension in valve spring

P_s = supply pressure

P_B = balancing air pressure

ΔF = force dead zone

The foregoing equations show that for a constant supply pressure and a given balancing pressure, the applied force can change by an amount

$$P_s A_I + F_0 - P_B[A_I - A_E] \dots [4]$$

without changing the balancing pressure. Further, it is seen that this dead zone is not a constant but consists of a constant minus an effective area imbalance. The position assumed by the valve in equilibrium is dependent upon the relative sealing of inlet and exhaust ports. If the inlet port will seal more perfectly than the exhaust port, it is necessary for the inlet port to open slightly for a constant balancing pressure to be maintained; hence force Equation [1] applies. Also, if the exhaust port seals more per-

⁸ Director of Research, Taylor Instrument Companies, Rochester, N. Y.

⁹ Engineering Research Division, Taylor Instrument Companies.

fectly than the inlet port, the exhaust port must remain slightly open and force Equation [2] applies. As the result of wear, oil film, and dirt from the air supply, or slight changes in alignment of the valve pin, it is possible for the balancing pressure to come to rest at either extreme of the dead zone. Unless a fixed bleed is introduced so as always to require one port to remain slightly open, it is impossible to predict with certainty which port will be slightly open at the equilibrium balancing pressure even for successive operations.

That such a dead zone actually exists in devices of this type has been determined experimentally numerous times by the writers by the following procedure: A constant loading force is applied to the diaphragm, and a constant flow of free air is metered into the balancing-pressure chamber from a separate pressure source. Air is bled to atmosphere from the chamber through an adjustable restriction and a flowmeter. The balancing pressure is measured with a manometer. By adjusting the restriction across a range, data can be obtained for a plot of balancing pressure versus net flow (either inlet or exhaust) to the chamber. It will be found that a change in the balancing pressure is required to cross from inlet to exhaust flow and that the magnitude can be predicted approximately by Equation [3] for this type of pilot-valve arrangement. Fig. 13 of this discussion

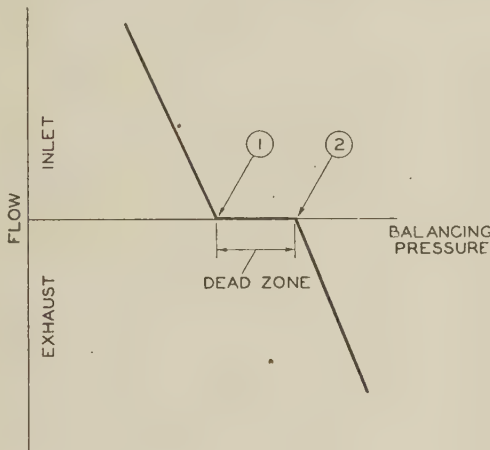


FIG. 13

shows the general characteristic of such a measurement where point (1) is the condition expressed by force Equation [1] and point (2) the condition expressed by force Equation [2].

When the analysis of the pilot-valve dead zone is applied to the apparatus in Fig. 12 of the paper, the possibility arises that some error in the instantaneous effective-area measurement can result unless a check is made to insure that the equilibrium position of the valve is either (1) or (2) for all readings. This is equivalent to saying that the "sensitivity" should be checked for constancy before each reading. A rough estimate indicates that if the sensitivity may be taken as 1 part in 50,000 of full load rating for the example in Fig. 11 of the paper, a definite error in instantaneous effective area will result if the pilot valve does not always assume the same position, either (1) or (2) in Fig. 13 herewith, for each reading.

The method of obtaining viscous damping is indeed very clever and, had the scope of paper permitted, a more detailed discussion of the damper and problems associated with various inertial loads would have been most interesting:

This paper is considered a fine contribution to the industrial-instruments literature and the authors and their company are

to be congratulated for furthering the development of force-measuring systems:

AUTHORS' CLOSURE

The discussion of Nichols and Woodhull amplifies the statements in the paper which deal with the effect of relay-valve parasitics on instrument accuracy and which state that existence of such parasitics means that a measuring diaphragm area of 3.5 sq in. is about the minimum for which an instrument of the type described may be expected to operate in the fractional percentage-accuracy class. Many excellent industrial instruments of a one-per cent-accuracy class are constructed with smaller diaphragm areas.

The mathematical analyses and experimental work described by the discussers are of course clear and correct as far as they go. However, the authors have had trouble in the interpretation of these results into terms of what Nichols and Woodhull call "dead zone and sensitivity." The authors therefore will redefine dead zone and sensitivity as used in the paper. Sensitivity is defined as the smallest increment of the measured force which will produce a reproducible change in the steady state instrument reading. It is not to be confused with "controller sensitivity." Dead zone is more difficult to define but the authors will define this term as inversely synonymous with sensitivity. Again the controller definition should not be used; the subtlety involved is that a dead zone is often deliberately introduced into certain controller designs for practical reasons at variance with concepts of instrument accuracy.

With this in mind, a satisfactory physical interpretation of the discussers' equations and experiments is not too difficult. Equation [1] is written "with the exhaust port closed and the inlet open an infinitesimal distance." This should be interpreted to mean the exhaust port seated but having leakage flow. With no exhaust leakage flow, the hypothesized condition would not correspond to a steady state, for the balancing pressure would continue to build up to a maximum. What is meant, therefore, is simply assumption of a condition in which the valve rests on the exhaust seat. Equation [2] is similarly interpreted as the inlet end seated, with leakage flow. Further, these assumptions are to be regarded only as setting limits. There is no necessary discontinuity between them. For example, the valve load may theoretically be shared in any proportion between the two ports with leakage flows in both.

Taking Equation [1] and applying a force increment to the left-hand side, it is seen immediately that the right-hand side can only equate by a change of balancing pressure because the areas are constants and a change in the small spring force F_0 is only brought about by valve motion and this connotes change in air pressure. Thus Equation [1] indicates zero dead zone and infinite sensitivity. Similarly Equation [2] indicates the same thing. The correct inference is not that sensitivity or dead zone is involved but that Equations [3] and [4] indicate the magnitude of possible instrument errors between the two cases. The discussers' ingenious experiment may be used to verify this magnitude. In the Thrustor design, the pilot-valve action is quite free of the ambiguities theoretically possible by Equation [4]. The experimental condition of the discussers in which an auxiliary air supply is introduced is only an experimental one since the actual instrument does not operate in this manner. The suggestion of introducing a positive air leak in the exhaust may not be without merit for some instruments using this valve principle. In the particular design of the paper, no great benefit is to be secured but on the other hand, the positional stiffness can be slightly impaired.

The stabilizer design might warrant a separate paper because it is, as the discussers point out, interesting in its simplicity as

an example of a damped servo system. Such a paper would no doubt be instructional as it would cover the behavior of a very simple mechanical system oscillating at damped resonant fre-

quencies. The lengthy treatment of the nonlinearities involved in the air spring and the relay valve precludes such an analysis in the present paper.

Apparatus for Analyzing Reservoir Fluids

By P. G. EXLINE¹ AND H. J. EN DEAN,² PITTSBURGH, PA.

In the petroleum industry the examination of subsurface samples of reservoir fluids has become a well-recognized procedure. Entirely satisfactory methods of sampling have been evolved. The authors have developed laboratory equipment designed to secure essential data with greater facility than hitherto has been possible. The apparatus is described and illustrated in some detail in the paper.

SINCE the pioneering work of Lindsly (1)³ in securing and examining subsurface samples of reservoir fluids, the recognition of the usefulness of this work has become widespread, and a substantial body of literature has been built up describing various techniques and applications. In this work, a reservoir fluid is understood to refer to the hydrocarbon liquids and gases as they exist in the producing formation. Samples of reservoir fluids may be secured by sampling with a specially designed bottom-hole sampler (1, 2, 3, 4) at the bottom of the well, or by taking samples of the liquid and gaseous phases in the proper ratios at the separator (5). Methods of securing suitable samples of either kind are described in the literature and require rather specialized techniques in order to secure representative samples. It is assumed without further discussion that with proper techniques satisfactory reservoir fluids may be placed in laboratory equipment for analysis whether the material was secured by bottom-hole sampling or by recombination of surface samples.

The data to be secured from the fluid include the initial saturation pressure or bubble point, the gas solubility in the oil at reservoir temperature and pressure and at intermediate pressures down to atmospheric, the per cent shrinkage of the liquid phase at reservoir temperature and pressure at various pressures down to atmospheric, the viscosity of the reservoir fluid as a function of pressure from reservoir pressure down to atmospheric pressure, the deviation factor of the liberated gases, the density of the fluid at reservoir pressure and temperature and at intermediate pressures.

The apparatus described in this paper was designed to perform these functions on reservoir fluids secured by both methods of sampling and to recombine the surface samples. In so far as was possible, the apparatus was designed to be highly flexible in operation to permit use of various analysis techniques. Four essential features common to equipment for this purpose include a pressure-tight system suitable for working pressures up to the maximum pressures, means for varying the volume of the system, means for stirring the contents of the system, and ability to operate at temperatures as high as 350 F. Various shapes and sizes of pressure systems have been used by other investigators, some of them being the sampler itself which was used to collect the sample

(1, 2, 3, 5, 6, 7, 8, 9). In the apparatus described, a rolling-ball viscometer is included as part of the pressure system.

The usual method of altering the volume of the system has been to introduce or withdraw mercury from it. Since mercury does not enter into combination with the reservoir fluids nor is it soluble with them, it can be considered as an inert material conveniently capable of altering the system volume. It has several drawbacks, as follows, which are desirable to avoid if possible: (a) Mercury is perhaps the most difficult fluid to hold in a pressure system since it can leak through crevices which are impervious to other liquids and gases. (b) Although no apparent reaction takes place with some reservoir fluids, notably the sour crudes of the Permian Basin, the mercury becomes emulsified with the oil, requiring considerable labor to clean the apparatus and to recover the mercury following a run. In the present apparatus, the use of mercury for changing the volume of the system has been avoided by use of a variable-volume cell in which a metal piston, operating through a packer, is used for altering the volume.

The fluids in the pressure system usually have been stirred by shaking or rotating the pressure chamber, in order to get the necessary mixing action of the fluid. Especially when mercury is present, the fluid in a long cylindrical chamber will be violently stirred as the mercury is allowed to flow from one end to the other. In the apparatus described, a reciprocating pump withdraws fluid from the bottom of the variable-volume cell, pumps it through the system to return at the top of the cell where it is violently jetted downward toward the gas-liquid interface. This pump is operated by a magnetic solenoid and hence needs no mechanical parts operating through a pressure seal.

THE COMPLETE PRESSURE SYSTEM

A schematic diagram of the complete pressure system is shown in Fig. 1. This system can be described as being made up of three independent parts as follows: (a) The analysis section in which both the bottom-hole samples and recombined samples are analyzed; (b) the gas-compression system used for the quantitative injection of gas into the analysis system for recombination with the separator liquid in the correct proportion; (c) the pressure-indicating system for indicating the pressure within either of the other two systems during a test. The analysis section consists of the variable-volume cell, the viscometer, the magnetic pump, and the necessary connecting tubing. The variable-volume cell and the viscometer are enclosed in an insulated cabinet containing equipment for heating and circulating the air to maintain constant temperature. During a test run, the analysis system is isolated from the other two sections.

The gas-compression system is composed of two pressure chambers and a high-pressure pump used for displacing mercury from one bomb into the other to compress the gas to be injected into the analysis section. The two reservoir bombs are identical with those used for the recombination experiments of Botset and Muskat (5).

The pressure-indicating system consists of two Bourdon-tube pressure gages, one of 2000 psi range, and the other of 7500 psi range. These two gages are connected to the system in parallel so that the lower-range gage may be isolated when operating at pressures above 2000 psi. A hand pressure pump, consisting simply of a small screw-operated piston working through a packer,

¹ President, Exline Engineering Company. Mem. ASME. Formerly, Section Engineer, Gulf Research & Development Company.
² Section Engineer, Gulf Research & Development Company. Mem. ASME.

³ Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Design Committee of the Instruments and Regulators Division and presented at the Spring Meeting, Tulsa, Okla., March 2-5, 1947, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of authors and not of the Society.

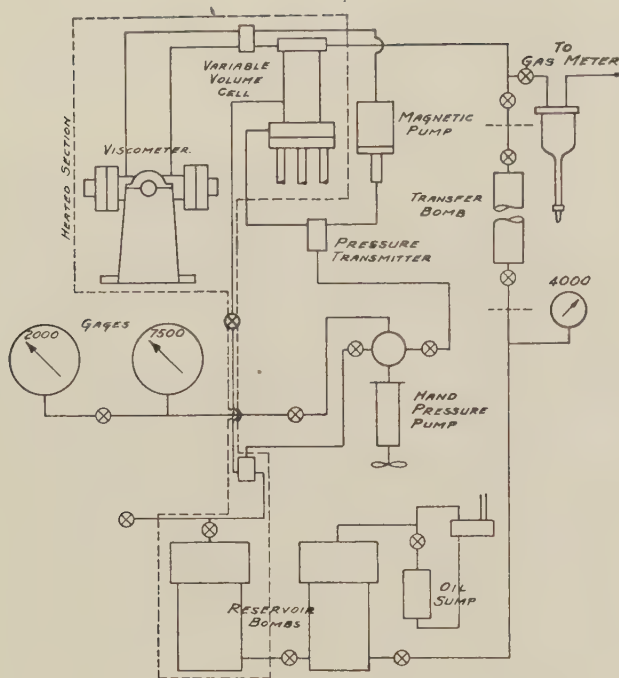


FIG. 1 COMPLETE PRESSURE-SYSTEM SCHEMATIC DIAGRAM

is used for taking up volume changes in the indicating system which result from changes in pressure. Two pressure transmitters are used for isolating the indicating system from the analysis and gas-compression systems, respectively. The transmitter consists of a small mercury U-tube with an electrical contact in one leg. During a test, the mercury level is maintained at the contact point by manipulating the hand pressure pump. The pressure-indicating system is completely filled with a light hydraulic oil, and the displacement of the hand pressure pump need be only that required to compensate for the compression of the oil and the expansion of the Bourdon tubes and other parts of the indi-

cating system. The contact of the pressure transmitter is connected with an electronic relay which causes a pilot light to glow when contact is made.

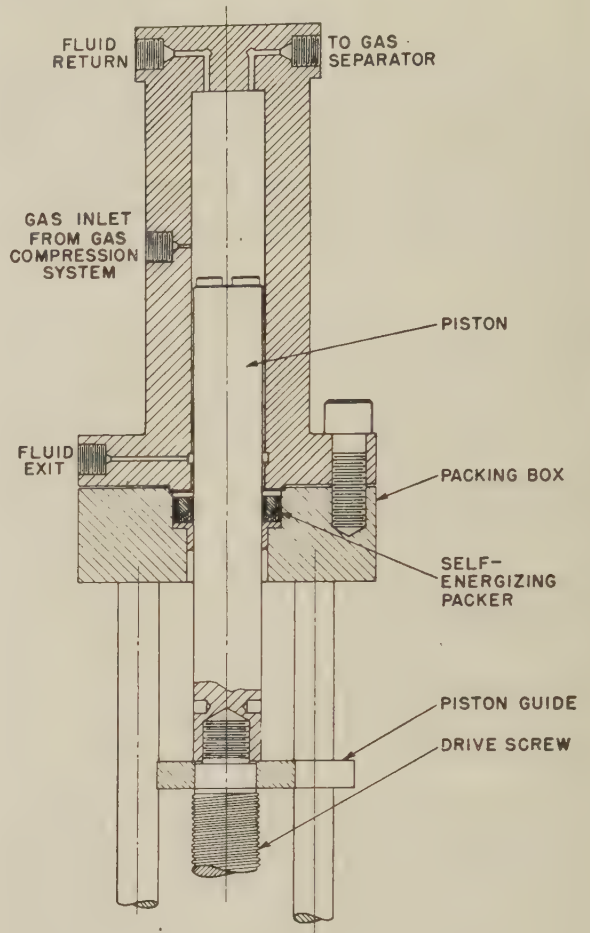


FIG. 2 VARIABLE-VOLUME CELL

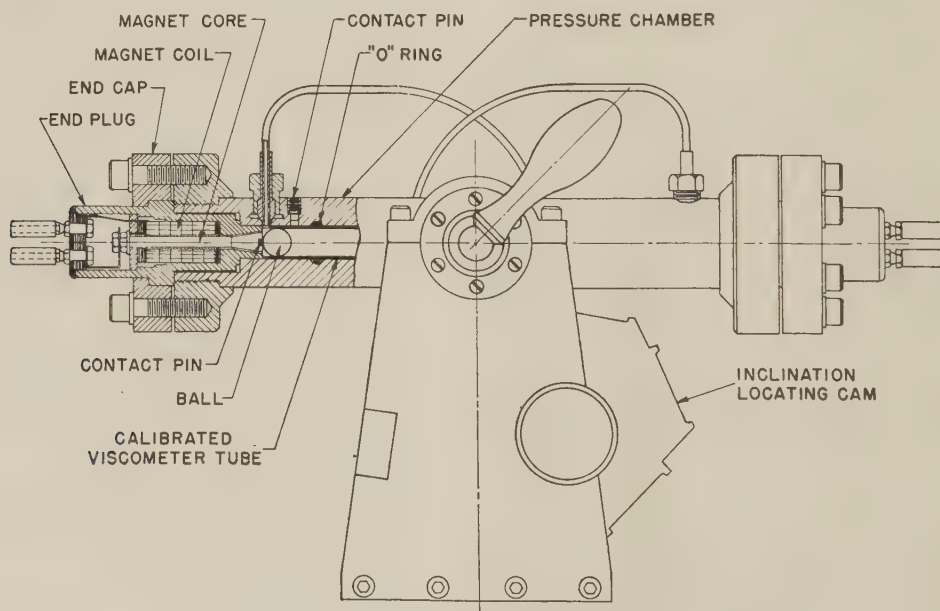


FIG. 3 ROLLING-BALL VISCOMETER

ANALYSIS SYSTEM

The principal components of the analysis system are the variable-volume cell, viscometer, and the magnetic pump which are shown in Figs. 2, 3, and 4, respectively.

Variable-Volume Cell. The variable-volume cell consists of a heavy-walled tubular piece closed at one end and bolted to a section containing the packer through which the piston enters the cell. The cell assembly is supported from a heavy base (not shown in Fig. 2), which carries the mechanism for moving the piston. The clearance between the piston and the wall of the cell is large enough to permit fluid to flow past it freely to the lower outlet of the cell. One opening at the top is used for completing the circuit through the other parts of the system while the other opening is used for withdrawing fluid from the system. A fourth opening at the center of the cell is used for injecting gas in the recombination experiments. Rotation of the piston is prevented by a guide working on diagonally opposite supporting posts. A heavy screw attached to the piston is given longitudinal motion by means of a rotating nut located between thrust bearings. Rotation of the nut is accomplished by means of a hand crank through a worm-and-gear combination.

A piston motion of 1 in. causes a change in volume of 20 cc, and the gearing is such that this travel is produced by 200 revolutions of the hand crank. Volume changes are indicated by means of a counter located directly above the hand crank, both of which may be seen on the right side of the panel in Fig. 7. The counter is coupled to the crankshaft by means of a Selsyn transmitter and a receiver. A magnetic locking device prevents the crank from turning when the Selsyn units are not energized. In practice, the counter is set to indicate the actual volume of the system for a given piston position so that, subsequently, the true value can always be read from the counter. The counter indicates volume to 0.01 cc.

The design of the packer offered the greatest difficulty of any

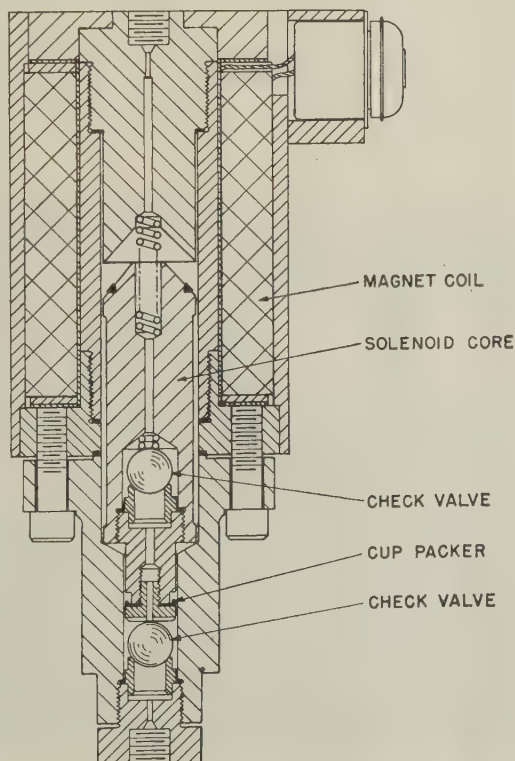


FIG. 4 MAGNETIC PUMP

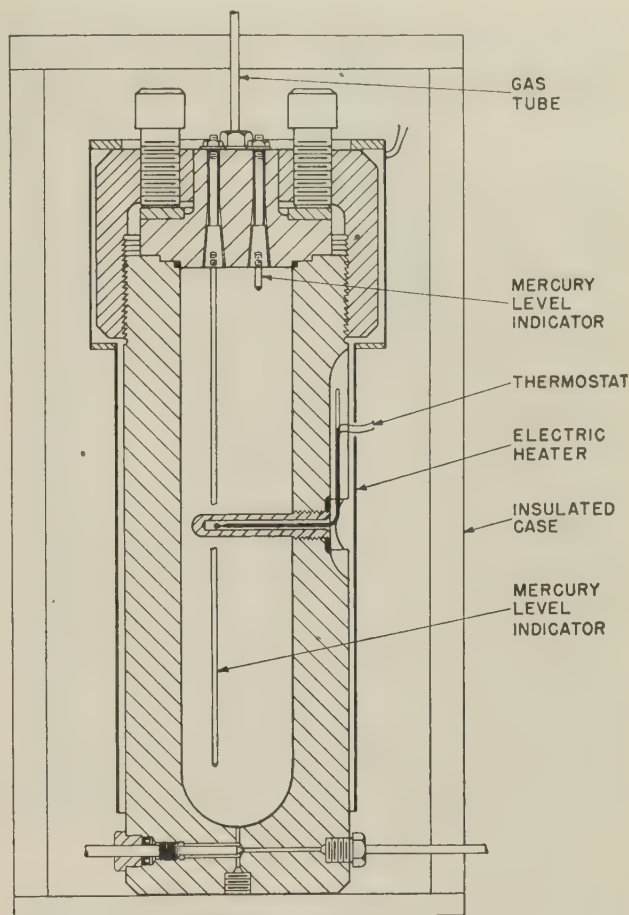


FIG. 5 RESERVOIR BOMB

component in the variable-volume cell. The requirements for the packer are very stringent in that it must be absolutely leak-proof for both gas and oil at pressures up to 6000 psi, and temperatures as high as 350 F. Tests of commercial packing arrangements and compounds soon showed that they would be entirely inadequate for such service. The solution of the problem was found in the self-energizing packing having a U-cup section. This design was found to be satisfactory at low temperatures, and any one of a number of different synthetic-rubber compounds was used. The maximum temperature at which the best of these compounds could be used was approximately 235 F. Above this temperature, the material tended to re-form and to vulcanize to the piston and to the packing compartment. The only material found to withstand higher temperatures was "Teflon," a recent du Pont development. While this material has met successfully all the temperature and pressure requirements, it does not have the resiliency of the synthetic-rubber compounds and requires extreme care in its manufacture and use. It has been our practice to use neoprene packers where temperatures do not exceed 200 F since their useful life is longer than that of Teflon.

Rolling-Ball Viscometer. The rolling-ball viscometer shown in Fig. 3 is a modification of that previously described by Exline and En Dean (10). Design modifications have been incorporated in the present instrument to secure greater reliability and trouble-free operation.

The instrument consists essentially of a tubular pressure chamber containing a precision-bore glass tube through which a steel ball can roll with a known clearance. The ends of the pressure chamber are closed by end plugs, each containing a solenoid mag-

net whose core communicates with the interior of the chamber and is electrically insulated from the chamber. With the ball held in contact with one of the end plugs by the magnetic field, it completes an electrical contact from the end of the magnet, through the ball to the metalized end of the glass tube and thence to a spring contact in the wall of the chamber. Completion of this electrical circuit energizes an electronic relay which in turn controls a timing clock. The connection is such that the timing clock operates only when the contact circuit is open. In operation, the instrument is given a definite angular displacement by forcing one end of it against a cam which can be set to secure any one of four fixed angles. A switch is thrown de-energizing the upper magnet holding the ball which is then free to roll down the tube. As it leaves the contact, the electronic-relay circuit is opened and a timing clock is caused to start. When the ball traverses the length of the tube and strikes the lower contact, the clock is thereby stopped, giving the reading of the roll time.

The principal design change in the instrument proper is the use of a precision-bore glass tube as the viscometer tube rather than depending on the inner bore of the pressure chamber for this purpose. In the past, difficulty had been encountered in making a precision bore to duplicate another tube closely. Since a number of these units were to be constructed, it was felt desirable that they all have interchangeable parts and that the calibrations be as nearly identical as possible. Since a large number of glass tubes can be formed over a single mandrel, their diameters can be held very closely to the same dimension without excessive expense. In the present design, the bore of the pressure chamber is drilled somewhat larger than the outside diameter of the glass tube. In order to prevent leakage of fluid between the tube and the bore of the pressure chamber, an annular groove is machined to take a synthetic rubber O-ring which would fit closely over the tube, making a perfect seal.

Electrical conductivity from the ball to the tube is obtained by metalizing the inner, outer, and end surfaces of the glass tube for a distance of a little more than one half of the diameter of the ball from the end of the tube. This metalizing is a recent development of the Corning Glass Company and is carried out at its plant. A spring-loaded contact pin is located in the wall of the pressure chamber near either end so that the ball point presses

firmly against the metalized end of the glass tube, thus facilitating continuity of the electrical circuit.

Other modifications of the original instrument include removing the jacket for circulating the temperature-control liquid and the insulation. This was done as it was found an air bath was much more convenient and satisfactory for temperature control.

Fluid connections to the viscometer are located near each end plug so that fluid can be circulated through the viscometer without trapping gas bubbles. While the circulating pump is operating, the tube is kept in an inclined position with the inlet at the lower end.

Temperature Control. Fig. 6 shows the rear of the apparatus with the cover of the insulated cabinet removed. The variable-volume cell and the viscometer can be seen in the left portion of the cabinet which is separated from the plenum chamber to the blower by a baffle. The blower, which is driven by a motor mounted outside the cabinet, forces air into a duct containing a series of finned heaters, having a total capacity of 2500 w. The lower side of the duct contains a number of louvers for securing improved circulation of air in the chamber. An adjustable thermostat protrudes into the cabinet for controlling the temperature, and the bulb of a thermometer, used for indicating the temperature, can be seen directly above the thermostat. Improved temperature control is obtained by placing but a part of the heating load on the thermostat. The voltage to the remainder of the heaters can be controlled by a variable-voltage transformer. Although the air temperature quickly reaches that for which the thermostat is set, its low heat-transfer coefficient requires that 1 hr or more pass before the heavy metal parts of the test elements reach that temperature. Once the correct temperature of the apparatus has been attained, any temporary disturbances of the air temperature are not perceptible in the temperature of the variable-volume cell or viscometer.

Use of an air bath is preferable to an oil bath in that the parts are much more easily accessible, and higher temperatures can be reached without objectionable fumes or the danger of fire.

Magnetic Pump. In quantitative analyses of this type where pressure and volume changes are produced in a system containing liquid and gas in contact, it is essential that a means of stirring or agitation be provided to assure equilibrium between the two phases. When the testing is conducted with a pressure system comprising interconnected pressure vessels, the most practical

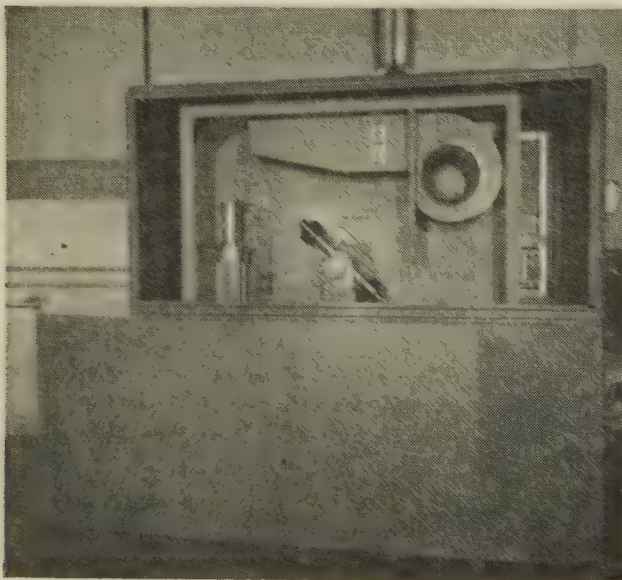


FIG. 6 REAR VIEW OF APPARATUS WITH COVER OF INSULATED CABINET REMOVED

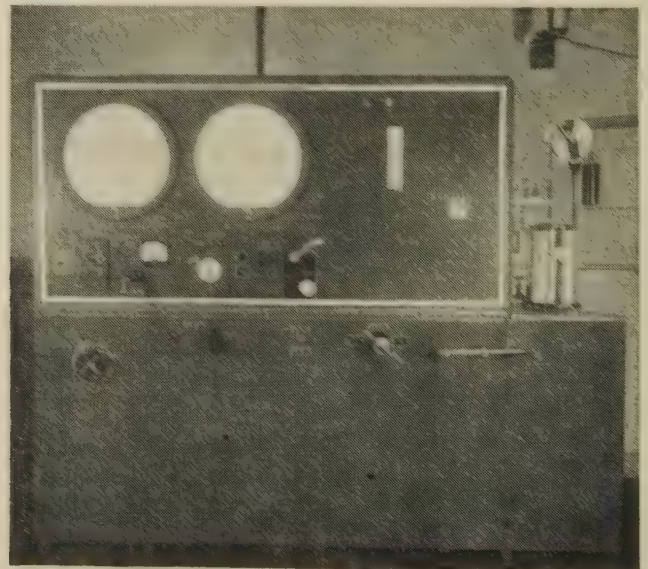


FIG. 7 CONTROL PANEL OF APPARATUS

method of obtaining equilibrium is that of circulating the test fluids. Any circulating device used must be absolutely pressure-tight and must cause no volume change of the system by its operation. The magnetic circulating pump shown in Fig. 4 was developed to fulfill these two requirements.

The pump is essentially a plunger pump with a traveling valve carried on a piston and a standing valve located in the pump body. The check valves are those commonly used in a $1\frac{1}{2}$ -in. oil-well pump with $\frac{1}{8}$ -in. balls. The piston is sealed by means of a small leather cup packing. The piston is attached to a soft iron bar which is pulled up into a conical pole piece when the solenoid winding, surrounding the pump body, is energized. This motion will pull fluid into the pump through the standing valve and at the same time force an equal volume of fluid out the upper connection to the pump. Upon de-energizing the magnet, the plunger will fall by gravity, causing fluid to pass through the traveling valve. The pump is driven by 100-volt direct current obtained by means of a commercial rectifier. The current is interrupted by means of a small motor-driven tip switch which interrupts the current at 2-sec intervals, the off time being substantially longer than the on time. The displacement of the pump is approximately 3 cc per stroke, and the energy is such that differential pressures as high as 300 psi can be built up. Each stroke is rapid enough that a pulse of fluid is sent around the system and jets into the variable-volume cell with considerable velocity, thus stirring the liquid contained there.

The magnetic pump is mounted outside the insulated cabinet to avoid overheating and damage to the winding. The volume of fluid contained in the pump is small, and the leads in and out of the cabinet are short so that the disturbance to the temperature of the system through operation of the pump is negligible.

GAS-COMPRESSION SYSTEM

The gas-compression system consists simply of two pressure chambers: one, the gas-compression chamber is shown in Fig. 5; the other, the reservoir bomb, is used for storing mercury which is displaced into the compression chamber to compress the gas.

The compression chamber is surrounded by a resistance-wire-wound heater and an insulating jacket. A thermometer well is welded into the wall of the chamber and projects across the interior. This holds a mercury-in-glass thermoregulator with a fixed setting of 212 F, which is used for controlling the temperature of the chamber. A controlled temperature is used for two purposes; as a known reference temperature to be used in computing the gas volumes, and to prevent condensation of any of the gases by liquefaction because present in the gas.

Two insulated contact pins are located in the cover plate of the compression chamber; one extending nearly to the bottom, and the other terminating just below the cover. These are used for locating the mercury-gas interface, and the volume contained between two horizontal planes just touching the terminal points of the contact pins must be accurately known.

The reservoir bomb is of identical construction, except that it does not have a thermometer well, being used at room temperature. Its cover contains but a single contact pin, terminating close to the top. It is used to indicate when the mercury reaches its highest permissible level in the bomb.

A small motor-driven reciprocating pump is used for pumping oil into the top of the reservoir bomb to displace the mercury.

APPARATUS

Tubing. All tubing connections are either $\frac{1}{4}$ in. OD \times $\frac{1}{16}$ in. ID, or $\frac{1}{4}$ in. OD \times $\frac{1}{32}$ in. ID. The heavier tubing is used wherever possible because of its rigidity. Where flexibility is required, as in the leads to the viscometer, the lighter tubing is used. Since the viscometer must be rotated through an angle of



FIG. 8 GASOMETER

over 90 deg, the leads are bent into spirals centered about the axis of rotation to secure sufficient length to keep the bending stresses down to a safe value.

All end connections are identical, the smaller tubing having a $\frac{1}{8}$ -in.-OD \times $\frac{1}{8}$ -in.-ID \times 2-in.-long sleeve welded over either end. The ends are turned to 60-deg cones and threaded with $\frac{1}{8}$ -in. 28 left-hand thread for $\frac{1}{2}$ in. A connector nut is slipped over the tubing and a small collar threaded on the end. When the connector nut is screwed into the female fitting, the conical end of the tubing is forced into a mating cone, providing a metal-to-metal seal of small area. This design of connection was first developed at the Fixed Nitrogen Laboratory and is now finding widespread use due to its reliability and simplicity. A joint may be made and broken many times without affecting its performance.

Gasometer. The gasometer, Fig. 8, for measuring the volumes of gas removed from the pressure has been described (3). It consists of a cylindrical holder having a re-entrant bottom which forms an annular space. This space is filled with mercury which acts as the sealing fluid. An inverted bell is suspended by a thin metal tape which passes over a pulley to a counterbalance weight. The side of the pulley toward the weight is cut in a spiral so that as the bell rises the weight moves farther from the center of the pulley, thus compensating for the decrease in buoyancy as the immersion in mercury decreases. A semicircular scale on the pulley indicates volume in co-operation with a vernier attached to the supporting post. The total capacity of the gasometer is 0.0160 cu ft, which can be read to 0.0001 cu ft by means of the vernier.

CALIBRATION

The volumes of all parts of the system must be accurately determined. If the piston of the variable cell is accurately made,

as it must be, this is a relatively simple process. The piston is retracted to its lowest position, the counter set to any arbitrary value V_1 , and the system filled with dry nitrogen to pressure P_1 , usually 200 or 300 psia. The piston is then run in an arbitrary amount ΔV , and the new pressure reading P_2 taken. Since nitrogen is not an ideal gas, its deviation from ideal behavior must be taken into consideration. If Z_1 is its deviation factor at P_1 and the temperature of the experiment, and Z_2 its deviation factor at P_2 and the same temperature, then

$$\frac{P_1 V_1}{Z_1} = \frac{P_2 V_2}{Z_2} \quad [1]$$

In this equation, V_1 is the total volume of the system with the piston in its initial position, and V_2 the volume with the piston in its final position. However, $V_1 - \Delta V = V_2$, which can be substituted in Equation [1] to give

$$\frac{P_1 V_1}{Z_2} = \frac{P_2 (V_1 - \Delta V)}{Z_1} \quad [2]$$

Solving this equation for V_1 gives

$$V_1 = \frac{\Delta V}{\frac{P_1 Z_1}{P_2 Z_2} - 1} \quad [3]$$

In practice, successive readings are taken while inserting the piston in stages to as high a pressure as desirable. Measurements are also taken while withdrawing the piston, stopping at the same values of pressure observed on the upward trip. A discrepancy in the counter readings when returning to the initial pressure is an indication either of leakage or of a small temperature change. Table 1 shows a set of data taken during an actual calibration run,

TABLE 1 VOLUME DETERMINATION OF VARIABLE-VOLUME CELL

P = observed pressures, absolute
 N = barometer pressure, 29.21 in. Hg
 Z = deviation factor of nitrogen at P
 ΔV = volume change
 V = calculated volume variable-volume cell

P obs., psia	Counter reading	ΔV	Z	V
243.75	997.10	2.90	0.9974	
843.75	951.70	45.93	0.9967	160.84
443.75	925.80	74.20	0.9961	160.84
343.75	909.85	90.15	0.9955	160.79
643.75	898.81	101.19	0.9954	160.89
743.75	890.77	109.23	0.9960	160.95
843.75	884.65	115.35	0.9960	160.94
943.75	879.80	120.20	0.9960	160.97
1043.75	875.93	124.07	0.9960	160.92
943.75	879.78	120.22	0.9960	160.99
843.75	884.36	115.44	0.9960	161.07
743.75	890.66	109.34	0.9960	161.12
643.75	898.64	101.36	0.9954	161.18
543.75	909.57	90.43	0.9955	161.30
443.75	925.47	74.53	0.9961	161.57
343.75	950.36	49.35	0.9967	162.28
243.75	996.51	3.49	0.9974	

and the computed volumes. In each case, the initial readings were used for the starting conditions, and the ΔV was taken as the total volume change to that point. The agreement between the volumes calculated at the different stops is unusually good, although the greater weight should be given to the determinations using the larger steps.

The only other significant calibration is that of the rolling-ball viscometer and this has been described elsewhere (10). This is done by observing the roll times when using fluids of known viscosity and density at atmospheric pressure. The assumption is made that any alteration of the dimensions of the glass tube and of the ball due to pressure is negligible. Previously, the calibration was shown as a series of curves with the absolute viscosity as ordinate plotted against $T(\rho_B - \rho_L)K$ as abscissa; where T is the

roll time in seconds, ρ_B the density of the ball, and ρ_L that of the liquid. A different curve was given for each angle of inclination of the tube. However, in that work, both ball and tube were of metals having substantially the same coefficient of thermal expansion. When a glass tube is used, the coefficients of expansion differ materially and a second parameter, temperature, is encountered.

CONCLUSION

Recent work by the senior author has indicated that it may be possible to construct the calibration of a rolling-ball viscometer from the dimensions of the apparatus alone. It is hoped that this can be made the subject of a future paper and that, for the sake of brevity, one of the results can be accepted without proof. This is that the velocity of the ball is proportional to the third power of the clearance between the ball and the tube. The experimental work has shown this to be true for the small changes of clearance caused by differential thermal expansion. As a result of this work, the calibrations are now expressed as a series of equations which include the temperature as a variable and the angle of inclination as a parameter. A group of such equations is shown in Table 2. The four positions indicate the four possible angles of inclination, and, for each angle, an equation is given for three sizes of balls. The tube diameter is 0.640 in.

TABLE 2 CONSTANTS FOR USE IN VISCOSITY COMPUTATIONS

Viscometer No. GLX-1049	
$\mu = \frac{T(\rho_B - \rho_L)K}{1000}$	
T = roll time, sec ρ_B = density of ball = 7.777 g per cc ρ_L = density of liquid, g per cc K = viscometer constant, as computed from equations given in Table t = temperature, deg F	
Tube No. 1075:	
$5/8$ + 0.006-in. ball....	Position 2 $K = 12.60 - 0.00896 t$ 3 $K = 8.70 - 0.00620 t$ 4 $K = 6.505 - 0.00465 t$
$3/8$ + 0.009-in. ball....	Position 2 $K = 5.775 - 0.00525 t$ 3 $K = 3.980 - 0.00360 t$ 4 $K = 2.980 - 0.00270 t$
$1/8$ + 0.012-in. ball....	Position 2 $K = 2.160 - 0.00270 t$ 3 $K = 1.482 - 0.00183 t$ 4 $K = 1.112 - 0.00138 t$
Tube No. 1078:	
$5/8$ + 0.006-in. ball....	Position 2 $K = 11.04 - 0.00820 t$ 3 $K = 7.615 - 0.00565 t$ 4 $K = 5.69 - 0.00420 t$
$3/8$ + 0.009-in. ball....	Position 2 $K = 4.84 - 0.00460 t$ 3 $K = 3.335 - 0.00315 t$ 4 $K = 2.50 - 0.00240 t$
$1/8$ + 0.012-in. ball....	Position 2 $K = 1.66 - 0.00220 t$ 3 $K = 1.147 - 0.00152 t$ 4 $K = .858 - 0.00114 t$
Tube No. 1105:	
$5/8$ = 0.006-in. ball....	Position 2 $K = 11.77 - 0.00860 t$ 3 $K = 8.12 - 0.00590 t$ 4 $K = 6.075 - 0.00445 t$
$3/8$ + 0.009-in. ball....	Position 2 $K = 5.27 - 0.00490 t$ 3 $K = 3.64 - 0.00340 t$ 4 $K = 2.71 - 0.00250 t$
$1/8$ + 0.012-in. ball....	Position 2 $K = 1.88 - 0.00240 t$ 3 $K = 1.295 - 0.00165 t$ 4 $K = .973 - 0.00125 t$

The method of introducing the sample into the pressure system needs no additional description here nor does the technique of recombining surface samples which has been adequately described by Botset and Muskat (6). The sole purpose of the paper has been to describe the design features of an apparatus which is composed of a number of elements, some new and some previously described.

BIBLIOGRAPHY

- "A Study of 'Bottom Hole' Samples of East Texas Crude Oil," by Ben E. Lindsly, R I 3212, Bureau of Mines, May, 1933.
- "A Bureau of Mines Study of a 'Bottom Hole' Sample From the Crescent Pool, Oklahoma," by Ben E. Lindsly, *The Petroleum Engineer*, vol. 7, no. 5, 1936, p. 34.
- "Practical Application of Petroleum Solubility and Libera-

tion Phenomena," by Ben E. Lindsly, *Oil and Gas Journal*, vol. 33, no. 36, 1935, p. 37.

2 "The Technique of Securing and Examining Subsurface Samples of Oil," by Ralph J. Schilthius, *Drilling and Production Practice*, 1934, p. 120; American Petroleum Institute, 1935.

3 "New Apparatus for Securing and Examining Subsurface Samples of Oil," by Paul G. Exline, *Drilling and Production Practice*, 1936, p. 126; American Petroleum Institute, 1937.

4 "Subsurface Sampling," by D. L. Katz, *Drilling and Production Practice*, 1938; American Petroleum Institute, 1939.

5 "The Study of Reservoir Fluids by Surface Recombination Experiments," by H. G. Botset and M. Muskat, *Journal of the Institute of Petroleum*, vol. 30, 1944, pp. 351-369.

6 "Equilibrium Cell for Investigating Properties of Fluids From Petroleum and Natural Gas Reservoirs," by C. K. Eilerts, R. V. Smith, and R. C. Wright, U. S. Bureau of Mines, RI 3514, 1940.

7 "Laboratory Technique and Equipment for the Study of Condensate," by W. F. Fulton, *Drilling and Production Practice*, 1938, p. 326; American Petroleum Institute, 1939.

8 "Phase Equilibria in Hydrocarbon Systems—I; Method and Apparatus," by B. H. Sage and W. N. Lacey, *Industrial and Engineering Chemistry*, vol. 26, 1934, p. 105.

9 "Apparatus for the Study of Pressure-Volume-Temperature Relations of Liquids and Gases," by B. H. Sage and W. N. Lacey, *Trans. AIME, Petroleum Division*, vol. 136, 1940, pp. 136-157.

10 "Viscosity Determination of Subsurface Samples of Crude Oil," by Paul G. Exline and H. J. EnDean, *Drilling and Production Practice*, 1939, pp. 659-665; American Petroleum Institute, 1940.

Discussion

W. E. ALLEN.⁴ The writer will deal mainly with the use of the apparatus in a laboratory doing considerable routine bottom-hole sample analyses. We have one of these units at the Tulsa laboratory of the Gulf Oil Corporation, and although it has been in operation for a very short time, many advantages can be seen over the apparatus which was used in the past.

Probably the greatest improvement is the elimination of the mercury from the analysis system. As the authors point out, mercury is difficult to hold in a pressure system. However, our main objection is its tendency to emulsify with the oil. The degree of emulsification is different with every sample analyzed, but we have never run an analysis without some emulsion being present. This always led us to question the accuracy of oil volumes measured with a metering mercury pump, with mercury emulsion in the system. Withdrawal of mercury from the system into the metering pump is bad practice because of the danger of getting some of the emulsion in the pump. The correction for the compressibility of the mercury is also eliminated from the volume calculations.

The determination of the viscosity of the oil in conjunction with the differential liberation analysis results in a considerable saving in time, and there is always a direct correlation between the viscosity and the rest of the analysis. Frequent checking of the "roll time" of the viscometer ball is a good means of determining when equilibrium has been reached within the analysis system.

The elimination of mercury from the analysis system plus the way in which it is calibrated makes it possible to tell by a glance at the counter dial, the exact volume of the system. This is helpful to the operator and permits him to make rough calculations as he progresses through the analysis. It also eliminates the old method of determining the saturation point and saturated volume of an oil. The usual method was to compress from a two-phase to a single-phase system or expand from a single-phase to a two-phase system by known increments of volume and record the pressure change. Plotting pressure change versus volume change gave two straight lines, the intercept of which gave the saturation point and the saturated oil volume. The variable-volume cell referred to in the paper permits a differential

liberation analysis to be just that after the initial saturation point is determined.

As previously mentioned, we have not had the unit a sufficient length of time to investigate fully all its possibilities in either routine analysis or reservoir work of a special nature. In spite of this, we are planning certain changes or improvements, whichever they may be called. The main purpose behind these changes is to make it possible for one man to operate the unit through the complete analysis. At present, two men are required at various times through the analysis and transfer procedure.

One addition planned is to incorporate a needle valve in the block containing the pressure transmitter and place it so that the valve seats on the analysis-system side of the mercury U-tube. At present it is important to keep a fairly low pressure differential across the U-tube, lest the mercury be blown out due to too high a differential. This is mainly true during the transfer process or charging the analysis system with a sample. By closing the proposed valve, the U-tube is taken out of the system which eliminates the necessity of the operator's trying to be in two places at the same time.

It is also impossible to evacuate the analysis system without pulling the mercury from the U-tube with the present hookup. Although it is unnecessary to evacuate the system prior to charging with a sample it may facilitate the cleaning operation following an analysis.

KENNETH EILERTS.⁵ There are two features of the apparatus described by the authors which have considerable novelty and, if they prove satisfactory in practice, will likely find application in equipment constructed for many other purposes. The use of an air bath for providing temperature control eliminates many of the inconveniences which attend the use of liquid baths. The U-cup packing which permits the use of a piston to change the volume of the equilibrium cell not only avoids the use of troublesome mercury for this purpose but provides for increased versatility in the design of variable-volume apparatus.

Our research at the Bureau of Mines on phase equilibria of natural hydrocarbon systems has involved P-V-T measurements in the temperature range, —100 to 280 F. Liquid baths are used to obtain the desired precision and uniformity in temperature control. However, to cover this temperature range, at least three bath liquids must be used so that they will have the necessary fluidity and yet be free from fire hazards. Connections to the apparatus and certain accessories are, of necessity, submerged in the bath liquid and are inaccessible until the bath liquid is drained away and perhaps even the bath compartment itself is removed. Valve stems, tubing connections, or other parts which must be passed through the bath wall to the apparatus, require packing glands in the bath wall that may leak and be a source of trouble, particularly in portable equipment.

These design difficulties would not be present in an air bath like that described by the author. It is necessary that the panels comprising the bath compartment be only heat-insulated, and a bath could be designed so that a panel or panels could be removed to expose the apparatus for service operations from any side or even from the bottom without undoing connections or destroying gaskets that would be essential if the bath held a liquid. If air or other gas baths can be constructed to provide adequate temperature control, the design and maintenance of certain constant-temperature apparatus will be greatly simplified.

Mercury is a useful material in handling and studying hydrocarbons under pressure, but it does seem to leak and get in the wrong places on the slightest provocation. When scattered on floors and tables, it is a health hazard and a source of annoyance

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to operators who like to practice good housekeeping. The piston for changing the volume of the equilibrium cell designed by the authors will help in avoiding some of the troubles associated with the use of mercury.

The piston makes possible an important design advantage. When mercury is used to vary the volume of a cell, it is not possible to locate in the bottom of the cell a window for viewing the phases or a valve for removing samples of the liquid phase, for example, because mercury pumped into the cell to provide a specific pressure may cover the window or valve. If the volume and pressure of a sample in the cell can be varied from the top by means of a piston, windows and valves can be located at some lower level in the cell with the prospect of their being in the correct location to study or sample satisfactorily a liquid phase obtained at any pressure.

This feature is particularly important in work with gas-condensate systems. The volume of the condensate varies considerably with pressure but, at reservoir temperatures, is only a small, fractional part of the total fluid (gas and liquid) volume. If the pressure on a fluid is varied between 5000 and 1000 psi by means of mercury, a shallow layer of condensate will float on the mercury surface as the surface is moved from some level near the top of the cell at high pressure to levels near the bottom of the cell at low pressures. The advantage of fitting a cell with a piston at the top would be that liquid hydrocarbons would collect and stay in one position at the bottom of the cell while pressure was being varied.

Other advantages are possible with a piston for varying the cell volume but all of them will depend on the serviceability of the packing which permits its movement. The ideal packing must permit movement of the piston and not leak gas at any of the pressures and temperatures of operation. The self-energizing, U-cup packing developed by the authors is placed in their cell so that it will be covered by a relatively viscous, lubricating, liquid phase, if one is present. However, this type of packing might also give satisfactory service if mounted in the top or at some other position in the cell where it will be in contact with only a gas phase.

AUTHORS' CLOSURE

The discussions contributed by Mr. Allen and Dr. Eilerts are particularly esteemed because of the wide background of experience in the analysis of reservoir fluids represented. Greater emphasis is given to the salient features of the apparatus sought in its development, i.e., facilitation of routine operation by the combination of various components into a single unit with simplification of controls, elimination of mercury in the analysis system, and the cleanliness and convenience of the air bath for temperature control.

The desirability of the isolating valve on the pressure transmitter described by Mr. Allen is recognized and the change has been incorporated in the apparatus. The original design of the transmitter used a steel ball floating on the mercury in the leg on the analysis side. A seat was provided at the top which could be closed by the ball as the mercury level rose to its maximum permissible height. However, the mercury thwarted our best efforts to produce a dependable tight seal at this point.

The apparatus described was designed solely for use with heavy-oil systems. It is believed entirely feasible to construct a variable-volume cell for use with gas-condensate systems as proposed by Dr. Eilerts. During the testing and calibration of the apparatus gas was held under pressure for relatively long periods of time without benefit of a liquid in contact with the packer. A point which would require investigation is the loss of hydrocarbon gases through the packer. The permeability of rubber and other elastomers to hydrocarbon gases has been investigated by Stross, Riley, and Eby⁶ who present quantitative results of their tests. A very rough estimate based on their data indicates that normal butane might be expected to pass through the neoprene packer at a rate of 0.05 ml per min from a system filled with gas at 5000 psi containing 10 per cent of butane. It is unlikely that such a loss would appreciably influence the tests but, if so, methods to combat it might be developed.

⁶ "Permeation of Hydrocarbon Gases Through Rubber Tubing". by M. J. Stross, J. F. Riley, and H. M. Eby, *Journal of the Institute of Petroleum*, vol. 30, June, 1944, pp. 153-166.

The 2000-Psi, 1050 F, and 1000 F Reheat Cycle at the Philip Sporn and Twin Branch Steam-Electric Stations

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The economical generation of steam-electric energy is affected by such factors as site, fuel, cycle, and equipment. A discussion of these basic factors serves as introduction to a description of the program involving the addition of 525,000 kw of generating capacity to the Central System of the American Gas and Electric Company at the Tidd, Philip Sporn, and Twin Branch Steam-Electric Stations. The discussion of the design employing 2000-psi 1050 F initial temperature with 1000 F reheat at the Philip Sporn Plant includes the location of the plant on the system, features of the site, the heat cycle, turbine, boiler, and plant layout. A net plant heat rate of 9270 Btu per net kwhr output is expected.

FUNDAMENTAL BASIS FOR ECONOMICAL GENERATION OF STEAM-ELECTRIC ENERGY

Introduction. The American Gas and Electric power systems, and more particularly the Central System, have almost from their inception been concerned with the economical generation of steam-electric energy as the foundation for a soundly conceived and successful electric-utility system. Elsewhere there have previously been described the developments at Philo (1),² where a 600-psi 725 F cycle with 725 F reheat was introduced in 1923; Deepwater (2), the first new station designed for 1250 psi operation; Logan (3), the first-planned postdepression major topping operation; Twin Branch No. 3 (4), the first successful 2500-psi installation in the United States; and Missouri Avenue No. 7 (5), the first commercial 1000 F unit. In that cycle there definitely also belongs Tidd Unit No. 1 (6, 7, 8, 9, 10), which followed Twin Branch No. 3, but which because of wartime exigencies could not take advantage of Twin Branch No. 3 experience and thus advance on that. Tidd No. 2, now under construction, likewise could not be designed as a forward step in generation technology—the need to get capacity on the line in the minimum possible period made substantial duplication almost mandatory.

But with the gain of a breathing spell brought about by construction of Tidd No. 2 and with the greater freedom to plan new facilities in equipment that the postwar period brought in, it became possible to try again to move forward on the front of economical energy generation by steam-electric processes. This, it is believed, has been at least partially accomplished successfully by the installations at Philip Sporn (Nos. 1 and 2) and at Twin Branch (No. 5) Stations. In each of these cases effort has

been directed to combine in optimum fashion the elements of site, cycle, fuel, and equipment. Each of these fundamental elements warrants further discussion.

Site Considerations. Offhand, it would appear that the selection of a site adequate for a power-station development is not an involved affair and that adequacy from the standpoint of area, condensing water, and location above flood plane are perhaps all one has to be concerned about. But this is a superficial view, for besides the factors enumerated, the following should also be considered: Location relative to present and future load centers and power flow on transmission lines; foundation conditions; amount of reasonably flat area available without moving roads or railroads, or moving large quantities of earth; railroad connections; transmission requirements for handling output; availability of economical coal supply; and ash-disposal facilities. Perhaps these are merely subsidiary themes to the main theme—location, but they are so important that they can be treated independently. There are, besides, lesser requirements for an ideal site. These are housing facilities for operating personnel, and tax costs. But the enumeration of the foregoing ought to be enough to indicate clearly the nature of the problem and to show why an exploration of a hundred miles or so of a typical river will frequently disclose, instead of the expected score or more of favorable locations, not more than perhaps two or three first-rate plant sites.

Fuel. As pointed out in the foregoing, fuel is a most essential element of site analysis and consideration. Yet fuel itself has an independent basis for study and examination. It should be apparent, for example, that a fundamental for developing the most economical energy supply is the need to exploit the lowest over-all fuel cost. This, however, does not mean necessarily the lowest-cost fuel in every case, and certainly not the lowest-cost fuel at the mine. It might, for example, definitely indicate the necessity for exploiting a high-quality fuel at locations at considerable distances from the mine; conversely, it might indicate the need, if the plant were located close to the mining regions, of exploiting a fuel of the very lowest quality—perhaps one that would have no commercial value if freight had to be added to its cost at mine. But this, in turn, raises the inevitable problem of making a determination as to whether to locate a power plant as close to the source of fuel as possible which will make lower fuel cost available but which will have added transmission cost. In contrast, it might be at some distance from the mine with higher transportation cost but lower electricity transmission cost. No answer will of course be attempted in this paper to this basic problem (11), but this needs to be emphasized: That the cost of fuel represents in most cases something between 70 and 85 per cent of the total production cost of energy at the bus bars. It seems inconceivable that a plant can be designed soundly without a thorough exploration of the fuel problem and a determination not only of what is available in the way of fuels, or what might be made available, but also without a study and determination of the probable trend of fuel costs over a reasonable period into the future.

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² Numbers in parentheses refer to the Bibliography at the end of the paper.

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Equipment. The type of equipment best suited for a particular development would obviously depend primarily on the system for which it is going to furnish the energy. For example, such questions as the size of unit, the pressure and temperature employed, and the cycle chosen will depend to a considerable extent on the fuel that can be developed, on the projected trend of fuel cost, and on the system and therefore to a considerable extent also on planned load factor. But there are still other factors that are bound to influence the choice of equipment. For example, if the system in question has a large proportion of industrial load and if it is the intention of those responsible for the policies of the company to continue to foster and develop such load, such a conviction must be reflected in the equipment in the power plant and the efficiency with which such equipment can produce electric energy. After all, it is inconceivable that central-station service can compete with isolated generation in the long run if central-station service is not on at least a par with performance being obtained by industrial plants. If such thinking is projected to its logical conclusion, and if recognition is given to the fact that there is no way of keeping from industry the benefits from developments by electric-utility systems of economies in generation, it follows that such advantages that a utility system derives from advances in the art of generation can be maintained only by continuing to be in the forefront of progress in the art.

It is pertinent, perhaps, to make the observation here that there has been a considerable amount of questioning of economies to be obtained from the successful development on a commercial scale of generation at higher pressures and temperatures, and therefore at higher thermal efficiency. Unquestionably, thermal efficiency alone can never be a proper objective in a successful commercial operation. But the developments in power generation over the last quarter of a century at least have clearly shown the economic benefit of more efficient generation. Looking back historically, it is now clear that many of those who have questioned the economics of more efficient generation have merely been slow in acknowledging the economic benefits that have flowed from such new developments. It is true that in holding back, many have been able to take advantage of the development work carried out and the concomitant headaches borne by others. But such a course can hardly be recommended as part of a program and philosophy of advancing technology to the end of achieving economic gain.

GENERATION CAPACITY ADDITIONS ON CENTRAL SYSTEM OF AMERICAN GAS AND ELECTRIC COMPANY

Basis for Capacity Additions. The principal or basic reason for the determination to proceed with the capacity additions discussed here was growth of load, or expected growth. This is the predominant reason for most capacity-addition projects and therefore presents nothing new. What is new, however, is judging the time factor—the time when it was expected the new capacity would be needed. When, early in 1946, the load and capacity situations on the system were reviewed, it seemed to be quite clear that no additional capacity would be required until sometime in the fall of 1949. This judgment seemed to be strongly corroborated by the fact that a large addition of capacity—100,000-kw Tidd Unit No. 1—had been brought on the line in September, 1945, at a period past the time when the all-time war peak load had been met. However, by early spring of 1946 it was apparent that the expected contraction of load in the post V-J period had been overestimated and that new capacity would be required by the early summer of 1948. To meet this schedule, the addition would have to be made at a location where such speed would be possible. This, in turn, pointed to a location where material and equipment specifications were already available.

Hence the decision to place the next capacity addition at Tidd. The fact that the only necessary transmission capacity for delivering it to the system could be provided by a simple double-circuiting operation of an existing double-circuit line between Tidd and Canton, Ohio, helped in this decision.

Subsequent further studies of the load and capacity situation on the Central System led to further revisions upward of the expected loads to be met in the succeeding four years. This, coupled with the further lengthening out of deliveries, led to the decision to proceed with the installation of Twin Branch Unit No. 5 and Philip Sporn Units Nos. 1 and 2. It was expected that with the placing into service of the last of these units, all the requirements for load on the Central System through the winter of 1950–1951 would be met.

Program of Capacity Additions, Their Location, and Transmission Connections. The program finally developed consists of the blocks of capacity additions as outlined in Table 1.

TABLE 1

Location	Nominal	Capacity Feeder capability	Date of completion
Tidd No. 2	100,000 kw	112,500 kw	August, 1948
Twin Branch No. 5	125,000 kw	137,500 kw	April, 1949
Philip Sporn No. 1	125,000 kw	137,500 kw	July, 1949
Philip Sporn No. 2	125,000 kw	137,500 kw	July, 1950

The location of this capacity on the system, including some of the new transmission facilities required to bring it into the system and deliver it to load centers, is shown on the map, Fig. 1. It has already been indicated that the only transmission change involved in connecting Tidd Unit No. 2 to the system is the double-circuiting of an existing line between Tidd and Canton. In Table 2 are shown all the additional transmission lines required to place the entire block of 525,000-kw capacity on the system.

TABLE 2

Line	Voltage	Tower miles	Circuit mile
Twin Branch-Fort Wayne	132 kv	82	82
Tidd-Torrey	132 kv	6	56
Sporn-Portsmouth	132 kv	65	65
Sporn-Philo	132 kv	7	14
Sporn-Turner	132 kv	7	14
Sporn-South Point	132 kv	15	30

This represents a total estimated cost of \$4,410,000 or a total cost per kilowatt net capacity involving the entire program, including Tidd, of \$8.40.

CAPACITY ADDITION AT TIDD

The subject of this paper being the 2000-psi, 1050 F, 1000 F reheat additions at Twin Branch and Philip Sporn, no attempt will be made to give an engineering discussion of the Tidd Station. The original Tidd Plant and the Unit No. 1 installation have been described elsewhere (6, 7, 8, 9, 10). The plant was definitely a wartime development and the engineering reflects to a considerable extent the limitations that power-plant designers and erectors were subject to during the wartime period. Nevertheless, Tidd No. 1 was designed to emphasize low capacity cost and economy in operation and has, it is believed, demonstrated advances in that direction. The plant is ideally located from the standpoint of load center, fuel, and water; the performance of Unit No. 1 has amply demonstrated these facts. In the planning of Unit No. 2 advantage was taken of that experience and a good many features of Unit No. 1 were duplicated. This is true of the turbine and of the heat cycle. But by the time Unit No. 2 came along, a considerable change had taken place in the price level of equipment and materials and therefore a number of major innovations were made to combat in so far as possible the rising

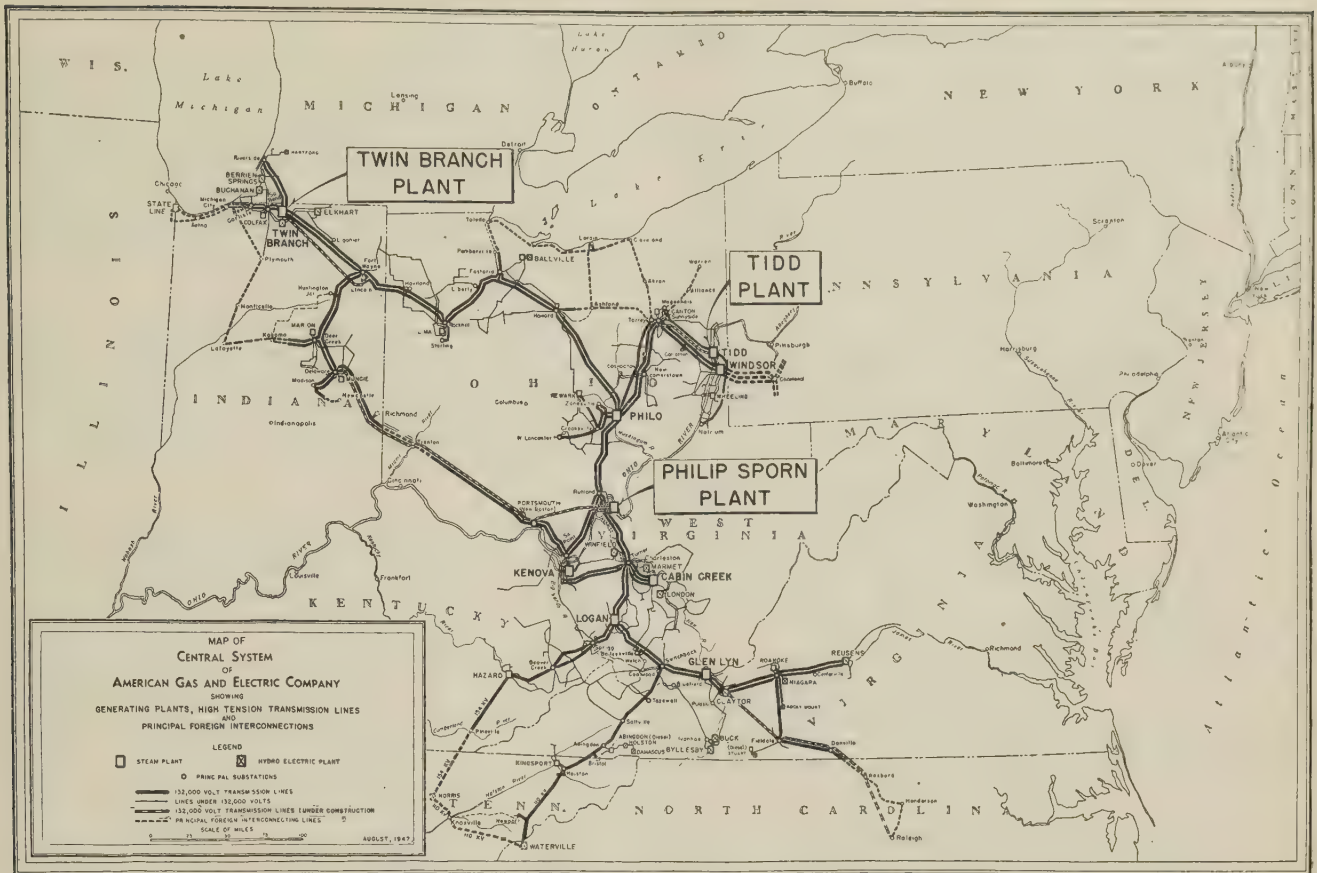


FIG. 1 DIAGRAM OF CENTRAL SYSTEM SHOWING INTEGRATED OPERATIONS OF THE INDIANA AND MICHIGAN ELECTRIC COMPANY, THE OHIO POWER COMPANY, APPALACHIAN ELECTRIC POWER COMPANY, WHEELING ELECTRIC COMPANY, KENTUCKY AND WEST VIRGINIA POWER COMPANY, INC., AND KINGSFORT UTILITIES, INC., BY A 132,000-VOLT TRANSMISSION SYSTEM

trend in costs. Included among these are the adoption of a single boiler for an output of 120,000 kw, a more shallow condenser pit, a totally different arrangement of step-up transformers, making possible more economical foundation construction and a lesser cable cost, and a number of other changes, all of which were of material help in counterbalancing to a considerable extent the rising cost factors. But because of the speed with which the job had to be brought on the line, no major change, either in cycle or in equipment, except in the boiler, was made from Unit No. 1, and the next major step in economical development of generation was left for Twin Branch Unit No. 5 and for the Philip Sporn Station.

CAPACITY ADDITION AT TWIN BRANCH STATION

Of all of the major stations on the Central System, Twin Branch is the most poorly located from the standpoint of fuel, that is, it is the farthest in location from the center of mining and it has therefore the highest freight rate. Nevertheless, the decision which was made to install another unit at Twin Branch was sound, it was concluded, for the following principal reasons: 1, The large saving in transmission investment that was possible; 2, it represented an incremental installation—perhaps the last increment that could be made at Twin Branch within the limits imposed by the circulating-water facilities; 3, the present addition of Unit No. 5 at Twin Branch fitted into a future program of developing an additional power-plant site in Indiana situated nearer local coal supplies, the development of the Philip Sporn Station, and the interconnection facilities between Ohio and Indiana.

Because Unit No. 5 at Twin Branch is similar from a standpoint of equipment, cycle, and basic arrangement, to the equipment, cycle, and arrangement followed at Philip Sporn, no space will be devoted to any discussion of these features at Twin Branch; the discussion later given of these features at Philip Sporn will apply almost equally well to the Twin Branch Station.

PHILIP SPORN STATION

Location and Organization. Fig. 1 shows the location of Philip Sporn Station on the Ohio River at the boundary of the two largest groups in the Central System—The Ohio Power Company and Appalachian Electric Power Company. Further, being located so close to the Portsmouth-Trenton-Muncie line, it is brought into juxtaposition to the system of Indiana and Michigan Electric Company. Thus the station could have been made a joint station for not only Ohio Power and Appalachian Electric Power, but for Indiana and Michigan Electric likewise. That a final decision was reached to make it a joint station of only the first two of these companies was due to a number of factors, among which are the balancing of the economics of locating a block of capacity at Twin Branch, and the decision to ultimately develop the additional new site in Indiana. But it needs to be emphasized that there is no basic physical or economic handicap to preclude such a triple-joint arrangement in the future.

From an ownership standpoint, the station will be so laid out that clear lines of demarcation of ownership can be established between the two parties to the joint operating agreement. This, however, does not mean that any duplication of facilities will be resorted to. On the contrary, every advantage will be taken

of the larger scale of operations a joint plant makes possible, to obtain all the economies for the two parties to the joint operating arrangement that could have been obtained in a single job of the same magnitude as the joint operation.

Site Features. The site itself consists of approximately 275 acres with 6000 ft of frontage on the Ohio River, see Fig. 2

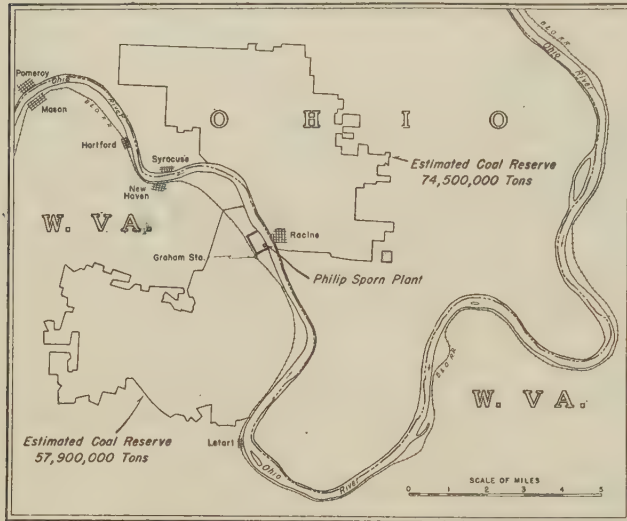


FIG. 2 MAP SHOWS RELATION OF 275-ACRE STATION SITE OF PHILIP SPORN STATION AND ITS CONTIGUOUS COAL RESERVES IN OHIO AND WEST VIRGINIA

Here, too, are shown the location of contiguous coal reserves in Ohio and in West Virginia amounting to approximately 22,340 acres and an estimated 132,000,000 tons of coal. This contiguity extends to the coal on the Ohio side because of the ownership of a sufficiently large block of connecting coal under the river bed of the Ohio River. The river at that point is not only navigable with 9 ft slack water navigation channel, but has a minimum flow of about 2200 cfs so that there is no question about the adequacy of the site for the development of a plant of at least three times the size of the present project. Furthermore, the coal reserves available give ample assurance of adequacy of economic coal supply with a minimum of transportation cost, both in the present and in the future, for a period extending over the depreciable life of the plant. The extent of that coal supply and its contiguity to the plant give further assurance that developments in underground gasification, if they should materialize, can be exploited in the future for such savings as may become possible. Thus the site seems to meet every requirement for economical generation discussed in the earlier part of the paper.

The Heat Cycle. The 2000-psi, 1050 F initial temperature, 1000 F reheat cycle that was adopted is shown in Fig. 3. In developing this cycle the major objective was optimum financial return, taking into consideration not merely economy of fuel but also reliability, for fuel economy means little to a plant that is shut down.

The heat rate at a net plant output of 141,000 kw is estimated at 9270 Btu per net kw-hr output with an expected 90 per cent boiler efficiency. This is a better heat rate than has been incorporated into any steam-electric station up to the present time.

In studying the economic possibilities of the various cycles, there was no question that a regenerative-reheat cycle offered the best heat performance. This led to comparative cost estimates of reheat versus nonreheat, with the balance in favor of reheat. As finally adopted, reheat gave roughly 5 per cent better heat rate, with a higher first cost of about 2 per cent. Needless to say,

this makes the reheat cycle for this station a most attractive investment.

Although the condenser is being designed for minimum condensate refrigeration and 0.03 cc per liter oxygen content at rated load, a deaerator to operate under positive pressure is being installed to insure thorough deaeration of the feedwater even under extreme load changes. In the past, the source of air in-leakage to condensers and other oxygen contamination of condensate has been difficult to track down completely, and the incremental cost of a deaerator over the equivalent closed heater is felt to be good insurance against boiler corrosion which, in turn, would adversely affect plant reliability.

Only six stages of bleed heating are used to achieve a final feedwater temperature of 441 F. Because of the large size of the installation, gland steam leak-off condensers and a separate evaporator condenser were found to be justified. Placing the heater drain pump on the 18th-stage heater was found more economical than on the 20th-stage heater which cascades its drips to the condenser.

The means used to cool the main generator is another example of the effort made to incorporate high reliability in the cycle. After leaving the condenser, a portion of the condensate is passed through the generator hydrogen coolers. The purpose is to preclude the frequent cleaning of the hydrogen coolers which would result if river water were used. After heavy rains the Ohio River is extremely turbid and the silt combines with leaves and twigs to plug tubes and small openings with a feltlike mat. It usually is satisfactory to use river water to cool equipment that may be cleaned without occasioning a plant shutdown, but with the particular type of hydrogen coolers available for the low-pressure generators, cleaning cannot be readily accomplished without losing generator capacity.

Although condensate temperatures are sufficiently low during winter months to cool the hydrogen and the generator, condensate temperatures are too high during the summer. A condensate cooler is therefore provided in which river water cools the condensate to the degree required in the summer. However, even after passing through the hydrogen coolers, the water is still cooler than in the main condensate stream. Therefore the cool condensate is sent to the suction of the hydrogen-cooler pump to pass again through the hydrogen cooler to condensate cooler to pump cycle. The cool condensate in this cycle "floats" on the main condensate stream but does not mix with it. Thus not only is the objective of reliability attained, but also fuel economy, since the main condensate stream is not cooled off by the cooler condensate in this subcycle.

Turbine and Its Features. The new cross-compound turbine is a new combination of high-pressure and low-pressure turbines. The 3600-rpm high-pressure element has a rating of 35,000 kw and a capability of 42,000 kw. Its throttle conditions are 2000 psig, 1050 F. The two emergency stop valves as well as the turbine inner shell will be chrome-nickel-molybdenum, columbium stabilized. The turbine outer shell will be molybdenum-vanadium.

The 1800-rpm low-pressure element has an intermediate-pressure turbine in tandem with a double-flow low-pressure section. The whole is rated at 95,000 kw with a capability of 108,000 kw, and its generator will be almost identical with those at Glen Lyn No. 5 and Tidd No. 1 and No. 2. The intermediate-pressure section receives steam from the reheater at 375 psig, 1000 F, and exhausts at about 8 psia to the double-flow low-pressure section. Its high-temperature parts will be of molybdenum-vanadium.

Among the features that improve the reliability and safety of these turbines are the following: Protection against sudden temperature changes by an initial-pressure regulator; protection

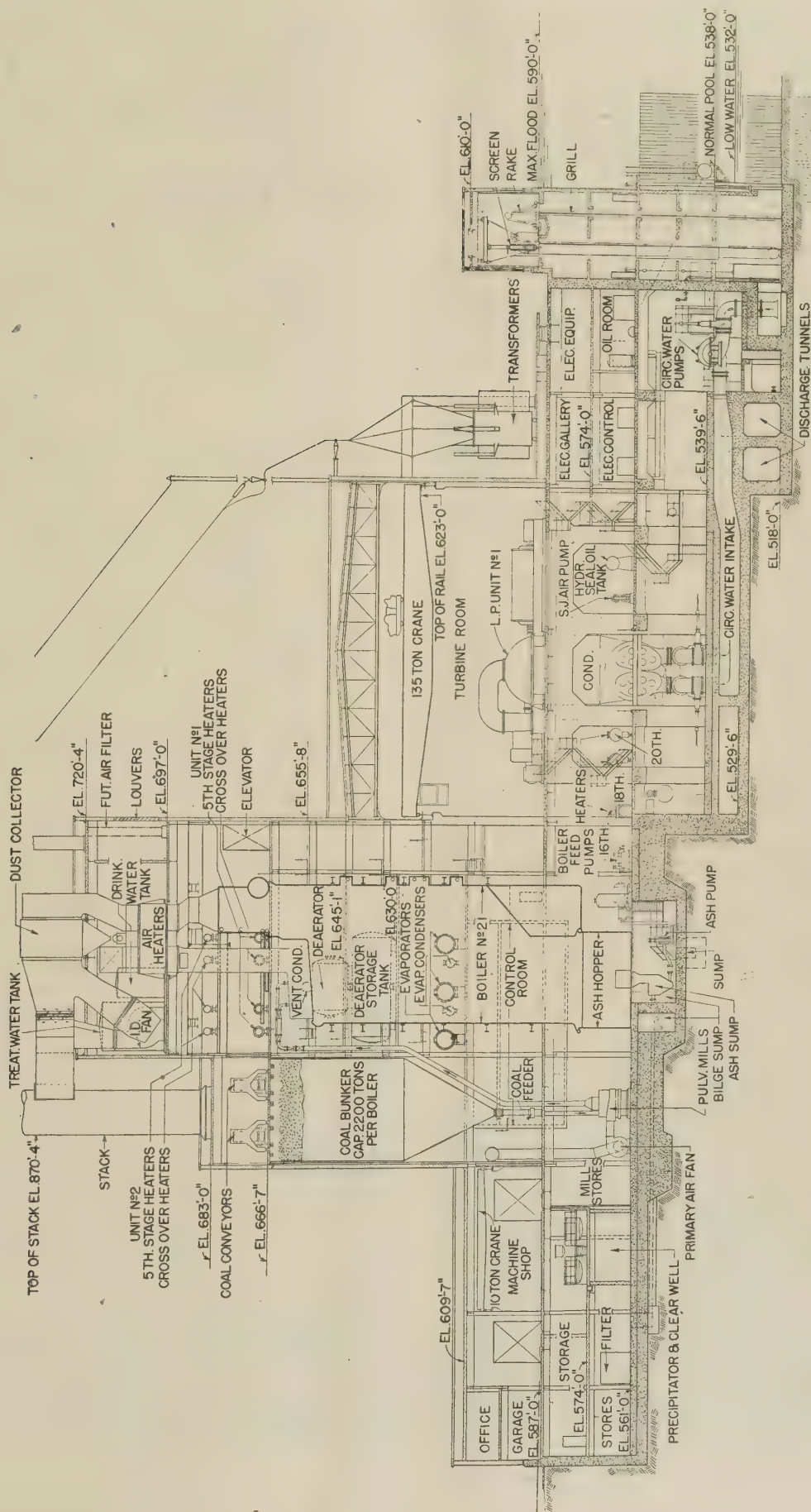


Fig. 3 THE PLANT CROSS SECTION SHOWS A COMPACT ARRANGEMENT IN THE INTERESTS OF SIMPLIFIED STATION OPERATION AND MINIMUM INVESTMENT

against excessive starting speeds by a two-speed, motor-operated, synchronizing device on the main operating governor; automatic steam-seal regulator and unloading valve; and automatic low-vacuum trip.

Boiler. The total heat input of 1,300,000,000 Btu per hr to produce 150,000 kw gross generation, makes the boiler one of the largest, if not the largest from the standpoint of net input that has been projected anywhere. It is designed for maximum continuous steam output of 935,000 lb per hr at 2035 psig, 1050 F at the superheater outlet. A temperature of 1050 F will be maintained between 850,000 to 925,000 lb per hr by an attemperator between the primary and secondary portions of the superheater.

After passing through the high-pressure turbine, the steam will be exhausted at about 400 psi, and some 850,000 lb per hr will be returned at 650 F to the reheater where it will be raised to 1000 F. This temperature will be maintained at 1000 F between flows of 750,000 and 850,000 lb per hr by an attemperator just before the inlet header to the reheater.

Flue gases discharged from the Ljungström air heaters go into the stack at about 235 F. The 12-in.-long cold-end layer of the air heater will be made of Cor Ten steel, which, together with recovery of air heated by boiler-room radiation losses, will resist corrosion caused by the low flue-gas temperature.

Five coal pulverizers will be furnished for each boiler. There will be ten burners per boiler of the multiple-intertube type. These fire vertically downward into the completely water-cooled tangent-tube furnace. The furnace is divided laterally by a curtain wall of bare tubes which serve to provide a substantial increment of cooling surface to realize reasonable heat-release rates. The curtain wall tubes are staggered in groups in the lower portion of the furnace to permit ready equalization of furnace pressures.

Another outstanding feature will be the attempt to run without induced-draft fans. To do this the entire casing will be made tight for a furnace pressure of 15 in. of water. However, induced-draft fans will be installed in case this forward step in the art of coal burning is not successful.

Electrical Arrangement. The electrical arrangement is shown diagrammatically in Fig. 4. The outstanding feature of the whole is its extreme simplicity: A 3-phase 52,500 and two 62,500-kva 3-phase transformers step up the high-pressure generator, and the two windings of the low-pressure generator, to 132,000 volts through a single switch to each of the two 132,000-volt buses. No low-voltage switches or buses are employed. Auxiliary power is furnished by two 6000/8000-kva transformers connected one to each of the two low-pressure generator windings and stepping down to 2300 volts. Seven feeders at 132,000 volts take the output of the station to the system. Each of these is switched through a single oil switch operable on either the main or the reserve bus.

Special Features. Prominent among factors that make for low-cost power generation is availability. Although it has been mentioned before in this paper, it can bear repeating that high thermal efficiency means little to a plant that is shut down. It is felt that availability must be inherent in the design of the equipment and its layout in the plant rather than merely providing a duplicate or a spare. True, availability must be correlated closely with plant investment as too much can be spent to avoid an outage. The designer keeps in mind that, after all, the plant is connected to a highly developed and reliable transmission system.

Being conscious that coal from strip mines must often be accepted, a straight down spout from the coal bunker to the pulverizer mill feeder will preclude hang-ups. To facilitate this layout no coal scales will be installed for individual boilers, but all incoming coal for the entire station will be weighed while on the belt.

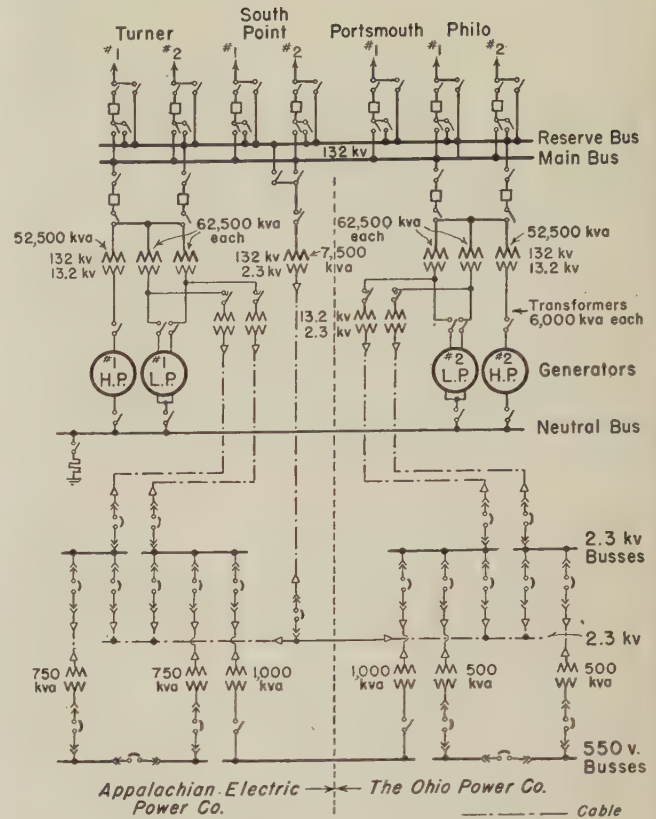


FIG. 4 ELECTRICAL ONE-LINE DIAGRAM

Four mills will ordinarily carry the load of one boiler; the fifth mill will act either as a spare to permit repairs on other mills, or may be called into service when the coal is of such poor quality that boiler capacity may be endangered.

It is trite to say that a clean plant is usually an efficient plant. It is likewise correct to say that a clean plant makes for low maintenance cost. This comes about in many ways, such as less cleaning of motors, relays, exposed threads on valves and last, but not least, the morale of the operating and maintenance force and their attitude toward the plant and their work. So it is natural that special effort be made to design for a clean plant in the many little design details in which this can be done. For example, a major source of dust and dirt within a plant is the expansion joints in the hot-air ducts between Ljungström air heaters and the burners. In the Philip Sporn plant all hot-air ducts under pressure will be welded, and bellows-type expansion joints will replace the usual slip-type joints.

Initially, outside air will be brought in on the windward river side of the plant at a point where the cleanest air should be obtained by taking advantage of prevailing winds. Such air will be distributed throughout the boiler room and condenser pit by two large ventilating ducts on each side of the elevator shaft. This clean air will be distributed throughout the boiler room to pick up heat and force dirty dust-laden air to the forced-draft fans, this action being aided by natural thermal air currents. Provision is being made to install air filters in the future, should these prove necessary.

Throughout the design, cognizance is taken that the plant operating organization has been brought down to the almost irreducible minimum, and that one of the major remaining problems of the designer is reduction of the maintenance organization which frequently has outgrown the operating organization in numbers. Reduction of maintenance organization automatically in-

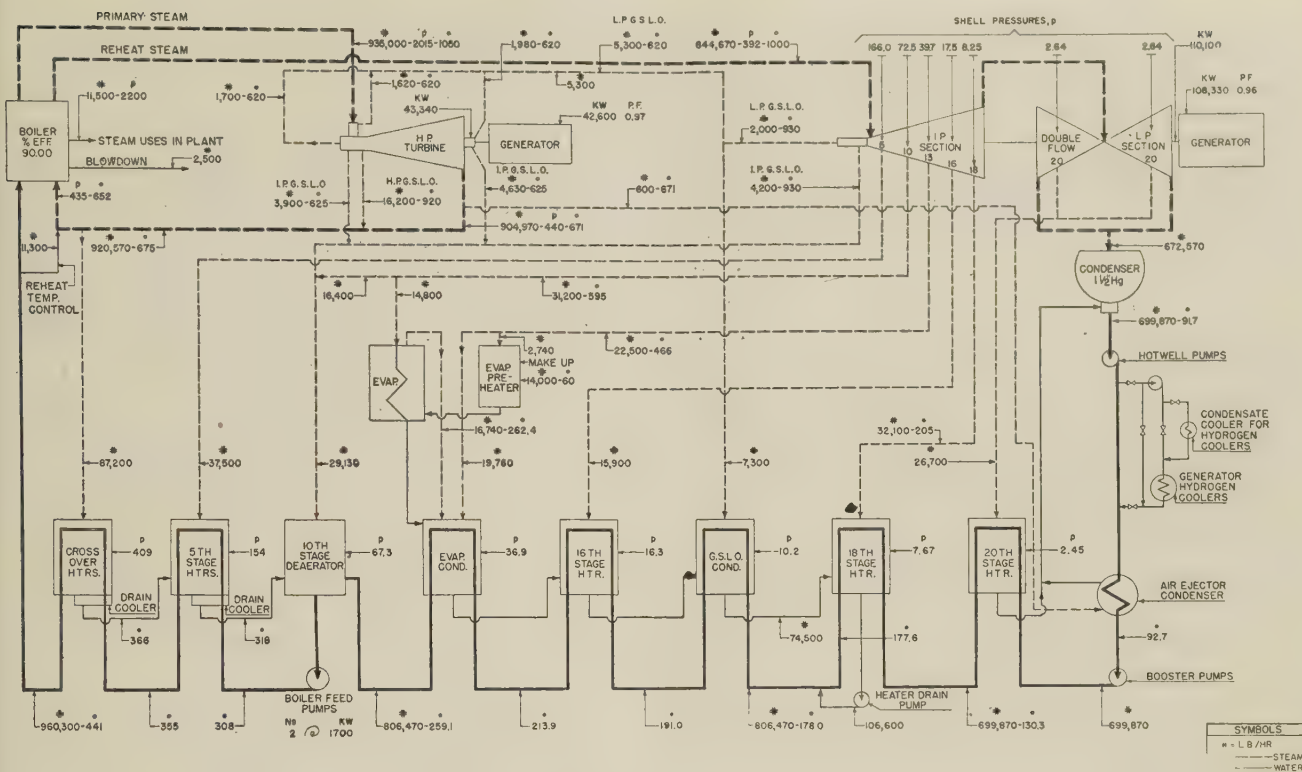


FIG. 5 THE NET PLANT HEAT RATE OF 9270 BTU PER KWH FOR A NET PLANT OUTPUT OF 141,520 KW IS BELIEVED TO BE THE BEST PERFORMANCE OF ANY STEAM-ELECTRIC STATION PROJECTED UP TO THE PRESENT TIME

volves optimum station availability and low maintenance cost. An outage of efficient equipment costs money by necessitating operation of submarginal equipment, besides the cost of fixing the failure.

The central control room that proved so successful at Tidd Unit No. 1 will be repeated at Philip Sporn as there is little doubt that the incremental cost of centralized control over decentralized control is warranted by the simplification of organization and reduction of outage time after trip-out.

Full condenser backwashing will be provided as the Ohio River becomes so extremely turbid after heavy rains. Long experience with other stations on that river has indicated that revolving screens are insufficient to cope with keeping condensers in operation under such conditions. It is possible to reverse the flow completely through each half of the condenser, thus removing the accumulation of leaves and leaf stems that become woven into a feltlike mat on the condenser tube sheet. Experience at the Windsor, Glen Lyn, Tidd, and Philo plants indicates no really satisfactory alternative to full-capacity backwashing.

Station Arrangement. The station cross section, Fig. 3, shows a compact and simple plant arrangement. For example, there are few plants in which the hot-air duct between air heater and furnace is as short or which so greatly simplifies the plant layout.

The steelwork is simplified by placing the stack over the coal bunker rather than over the boilers, making it possible to install the dust collector in a simple and economical layout.

As was the case at Tidd, the two center units of the ultimate six-unit plant are being installed to make it possible to place the office, machine shop, and storage facilities in the center of the plant, while the future extensions will be two units to the north side and another two units to the south.

Instead of an auxiliary bay between the boiler and the turbine, the auxiliaries will be placed between the two boilers. These

auxiliaries are bleed heaters, evaporators, deaerators, and boiler feed pumps.

As Fig. 3 also clearly shows, the main power transformers have been placed immediately behind the riverward wall of the turbine room. This not only makes for the shortest possible leads between generator and transformer, but makes possible also an ideal location for auxiliary transformers. Structurally, too, this provides a very low-cost support for the heavy weights involved.

The design of Tidd No. 2, Philip Sporn Nos. 1 and 2, and Twin Branch No. 5 is being handled by the American Gas and Electric Service Corporation which is also supervising all of the construction operations.

BIBLIOGRAPHY

- 1 "Philo Station Sets Record, General Conditions and Mechanical Features—I," by M. L. Sindeband and Philip Sporn, *Electrical World*, August 22, 1925.
- 2 "Philo Station Sets Record, Electrical Features and Control—II," by M. L. Sindeband and Philip Sporn, *Electrical World*, August 29, 1925.
- 3 "Deepwater—A New Idea in Central Stations," Anon., *Power Plant Engineering*, vol. 33, Nov. 15, 1929, pp. 1204-1214.
- 4 "Deepwater—A Double Purpose Station," Anon., *Electrical World*, vol. 93, May 11, 1929, pp. 919-923.
- 5 "Logan Steam Plant—A Landmark," by Philip Sporn, *Electrical World*, vol. 106, April 11, 1936, pp. 31-33, 98, 100.
- 6 "The Logan Steam Plant of Appalachian Electric Power Company," by Philip Sporn, *Southern Power Journal*, June, 1938.
- 7 "Logan Operating Experience," by Philip Sporn, *Trans. ASME*, vol. 60, July, 1938, pp. 421-422.
- 8 "Logan Operating Experiences," by Philip Sporn, *Trans. ASME*, vol. 162, April, 1940, pp. 253-256.
- 9 "Twin Branch Plant of Indiana & Michigan Electric Company (b) Twin Branch Extends High-Pressure Economies," by Philip Sporn, *Electrical World*, vol. 116, Oct. 18, 1941, p. 80.
- 10 "Twin Branch Plant of Indiana & Michigan Electric Company

(i) "Technically Sound and Commercially Workable," by Philip Sporn, *Electrical World*, vol. 116, Oct. 18, 1941, p. 80.

"The 2500-Psi Twin Branch Plant of Indiana & Michigan Electric Company (a) Operating History of the 2500 Psi Twin Branch Plant," by Philip Sporn and E. G. Bailey, *Trans. ASME*, vol. 66, 1944.

5 "One Thousand F—A New High in Steam Temperature" (Missouri Avenue Plant of Atlantic City Electric Co.), by Philip Sporn, *Electrical World*, vol. 126, Aug. 17, 1946, pp. 60-67.

6 "The Tidd Plant of the Ohio Power Company," by S. N. Fiala and E. H. Krieg, *Power Plant Engineering*, March, 1947.

7 "Steam Generation at the New Tidd Plant," by S. N. Fiala and L. B. Schueler, *Combustion*, vol. 18, October, 1946, pp. 30-35.

8 "Modern Steam Electric Power Station Built on Riverside Site," by H. A. Kammer, *Engineering News-Record*, vol. 137, Oct. 31, 1946, pp. 591-594.

9 "Tidd Generating Plant Electrical Features," by F. A. Lane, *Electrical World*, Aug. 16, 1947, and Aug. 30, 1947.

10 "Simple Tidd Plant Efficient and Economical," by Philip Sporn, *Electrical World*, Aug. 2, 1947.

11 "Cost of Generation of Electric Energy," by Philip Sporn, *Proceedings of the ASCE*, vol. 63, 1937, p. 1925.

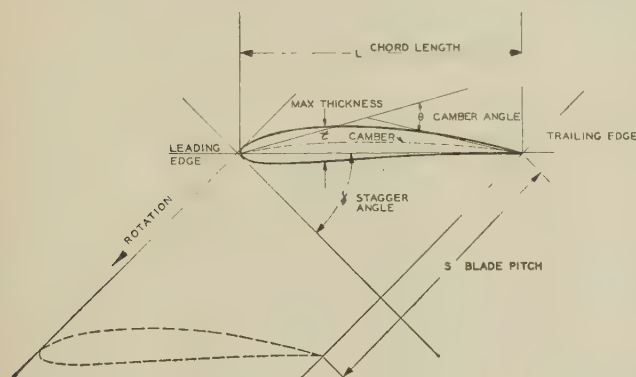
Axial-Flow Compressors for Gas Turbines

By A. I. PONOMAREFF,¹ LESTER, PA.

The development of the gas turbine during the last decade has opened a new field for compressor application. The compressor is an essential component of every gas-turbine power plant. It delivers air at some elevated pressure to the burner in sufficient quantity to maintain an efficient combustion of fuel and to cool the products of combustion to some acceptable temperature at the turbine inlet. The axial-flow compressor is particularly adaptable to gas-turbine applications for aviation, marine, or land service. This paper deals with the operating principles, characteristics, types, and other data pertaining to the axial-flow compressor.

NOMENCLATURE

The following nomenclature is used in the paper:



TYPICAL AIRFOIL BLADE SECTION AND TERMINOLOGY USED IN CONNECTION WITH BLADING OF AXIAL-FLOW COMPRESSOR

- C_m = mean relative velocity, fps
- C_L = lift coefficient
- k = ratio of specific heats
- L = blade-section chord, ft
- N = speed, rpm
- P_1 = inlet pressure, psia
- P_2 = intermediate pressure, psia
- P_3 = discharge pressure, psia
- Q = volumetric flow, cfm
- S = blade pitch, ft
- T = absolute temperature, deg R
- T_1 = absolute inlet temperature, deg R
- T_2 = absolute discharge temperature, deg R
- T_s = standard temperature, deg F
- U = blade speed, fps
- W = weight flow, lb per sec
- α = outlet-flow angle; between relative velocity leaving rotating row and axis of rotation

- β = inlet-flow angle; between relative velocity entering rotating row and axis of rotation
- ν = kinematic viscosity sq ft per sec

INTRODUCTION

When applied to a gas-turbine plant, the compressor must meet very rigid specifications. The compressor absorbs roughly about two thirds of the power developed by the turbine, and its efficiency therefore is of paramount importance. Fig. 1 shows the effect of the compressor efficiency on the over-all efficiency of an open-cycle gas turbine operating at various pressure ratios. The large amount of excess air necessary for cooling the products of combustion requires a large volume of air to be handled by the compressor even for a gas-turbine plant of relatively small output. For a marine application, the compressor must be light in weight and occupy small space, while for aviation service it must have a small frontal area as well.

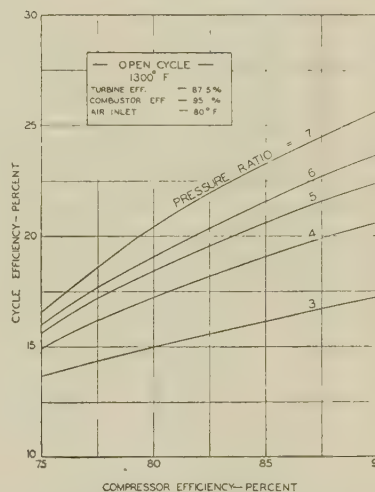


FIG. 1 EFFECT OF COMPRESSOR EFFICIENCY ON NONREGENERATIVE OPEN-CYCLE EFFICIENCY

The axial-flow compressor fits in exceptionally well as a component part of any gas-turbine power plant for aviation, marine, or land applications. It is a high-efficiency, high-capacity, and high-speed machine.

PRINCIPLES OF AXIAL-FLOW COMPRESSOR

An axial-flow compressor in construction, resembles the familiar reaction steam turbine. However, there is a material difference in the fundamental dynamics of the flow associated with these two machines. In a steam turbine the blade path is arranged for an expanding or accelerating flow; in a compressor, for a diffusing or retarding flow. In the expanding passage of a turbine the pressure decreases in the direction of the flow, and the boundary layer is continuously supplied with energy to accelerate the gas particles, which have been slowed down by friction, thus producing a stable flow. In a diffusing passage of a compressor, the flow is inherently unstable. With a negative pressure gradient, the pressure forces are acting in a direction opposite to the flow, and tend to retard further the gas particles in the boundary layer, producing eddying and backflow.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 47-A-28.

In an axial-flow compressor, air or gas flows in a general axial direction through a bladed annulus concentric with the axis of rotation. In the rotating row of blades, the air or gas is deflected through a small angle in the direction of rotation. This change in direction of the flow is accompanied by a decrease in relative velocity with resultant pressure rise through diffusion. Due to the unstable nature of the flow in a compressor, only a small change in direction of the flow or pressure rise is possible across a stage. Hence a large number of stages are required to obtain even a small compression ratio. The necessity for small turning angles across a stage compels the axial-flow-compressor designer to discard already developed steam-turbine blades and seek new aerodynamic sections.²

While the condensing steam turbine for the pressure ratio of 500:1 or over is in common usage, the pressure ratio above 7:1 is seldom developed in one compressor cylinder without resorting to intercooling.

Intercooling. In compressor applications involving relatively large pressure ratios, it is the usual practice to employ two or more compressors in series with intercoolers between. Intercooling between the compressor stages makes the compression approach somewhat the isothermal process, resulting in a considerable reduction in the power required. In a gas-turbine power plant, the saving in power required to drive the compressor due to the subdivision of compression and intercooling does not result in a corresponding improvement in the fuel rate or cycle efficiency. In a simple (no regeneration) open-cycle gas-turbine power plant, for instance, the improvement in cycle efficiency is so small that the introduction of an intercooler in many cases cannot be justified. In a regenerative cycle, a substantial improvement in cycle efficiency may be obtained through intercooled compression and it is a general practice to use at least one intercooler.

² "The Characteristics of 78 Related Airfoil Sections From Tests in the Variable-Density Wind Tunnel," by E. N. Jacobs, K. E. Ward, and R. M. Pinkerton, U. S. National Advisory Committee for Aeronautics, Report No. 460, 1933.

A convenient method of analyzing the effect of intercooling on both the work and efficiency of the over-all cycle is to analyze the small intercooled portion of the entire cycle as a separate cycle after which its work and efficiency may be added to the original cycle without intercooling, and the combined effect estimated.

Subdivision of compression ratio into two equal parts with intercooling after the first-stage compressor results in a minimum work of compression. In a gas-turbine power plant, however, such division of the compression does not result in the best cycle efficiency unless 100 per cent regeneration is employed. The best division of compression ratio for the maximum improvement in the cycle efficiency depends upon the over-all cycle efficiency and amount of regeneration employed. Fig. 2 shows the optimum ratio of the compression ratio of the high-pressure-stage to that

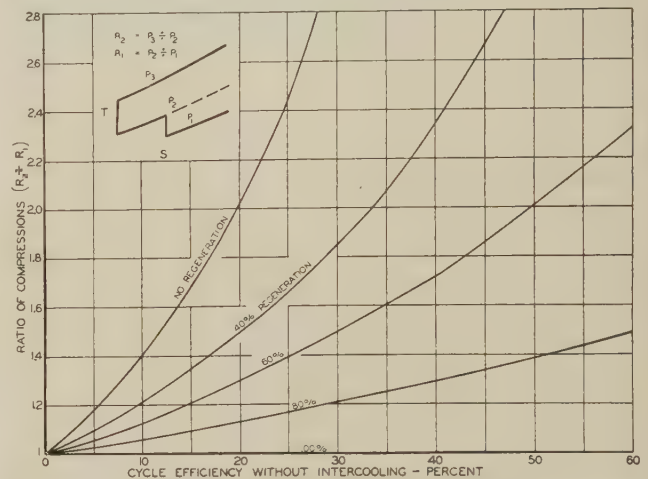


Fig. 2 IDEAL DIVISION OF COMPRESSOR WORK FOR INTERCOOLING (P_1 = initial pressure, abs; P_2 = intermediate pressure, abs; P_3 = final pressure, abs; R_1 = low-pressure cylinder pressure ratio; R_2 = high-pressure cylinder pressure ratio.)

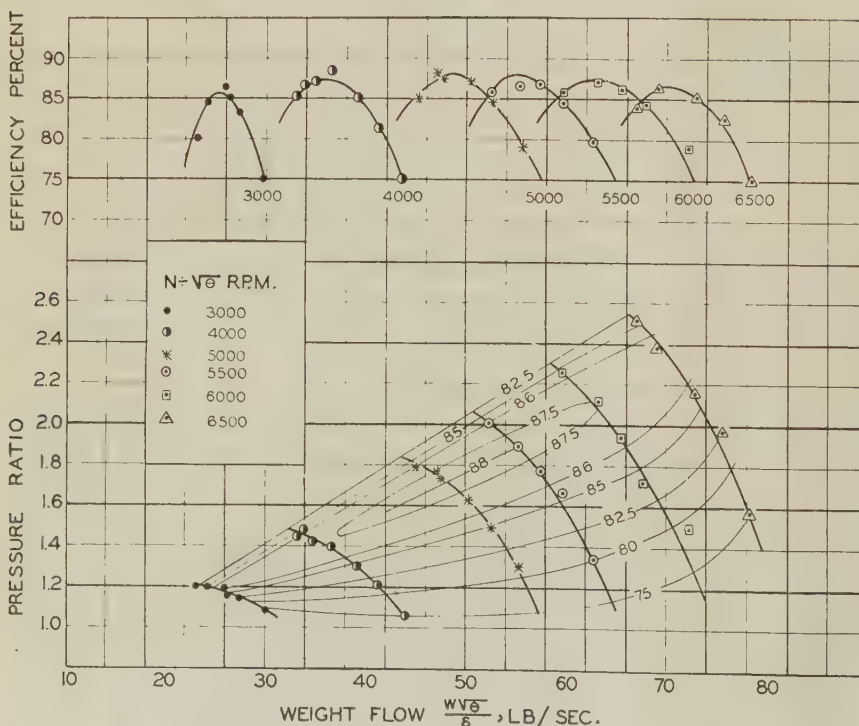


Fig. 3 CHARACTERISTICS OF 9-STAGE AXIAL-FLOW COMPRESSOR (Test results based upon 14.7 psia and 80 F inlet-air conditions.)

of the low-pressure-stage compressor for various cycle efficiencies and degrees of regeneration.

Characteristic Curves. For evaluation of the usefulness of a compressor, its performance is usually presented in the form of characteristic curves such as shown in Fig. 3 for the Westinghouse 9-stage axial-flow compressor, Fig. 4, installed in 1946 in the aviation gas turbine laboratory to supercharge another 24-stage axial-flow compressor, Fig. 5. The curves in Fig. 3 show the pressure ratio and efficiency versus the flow in pounds per second (volumetric inlet air flow in cubic feet per minute may be used also) for various test speeds from 3000 to 6500 rpm.

Unless some torque-measuring device is used for compressor tests, the efficiency is usually based upon the pressure and temperature measurements at the compressor inlet and discharge hoods, and an isentropic temperature rise, Fig. 6, for the corresponding pressure ratio and air-inlet temperature. This isentropic, sometimes called adiabatic, efficiency may be expressed as follows

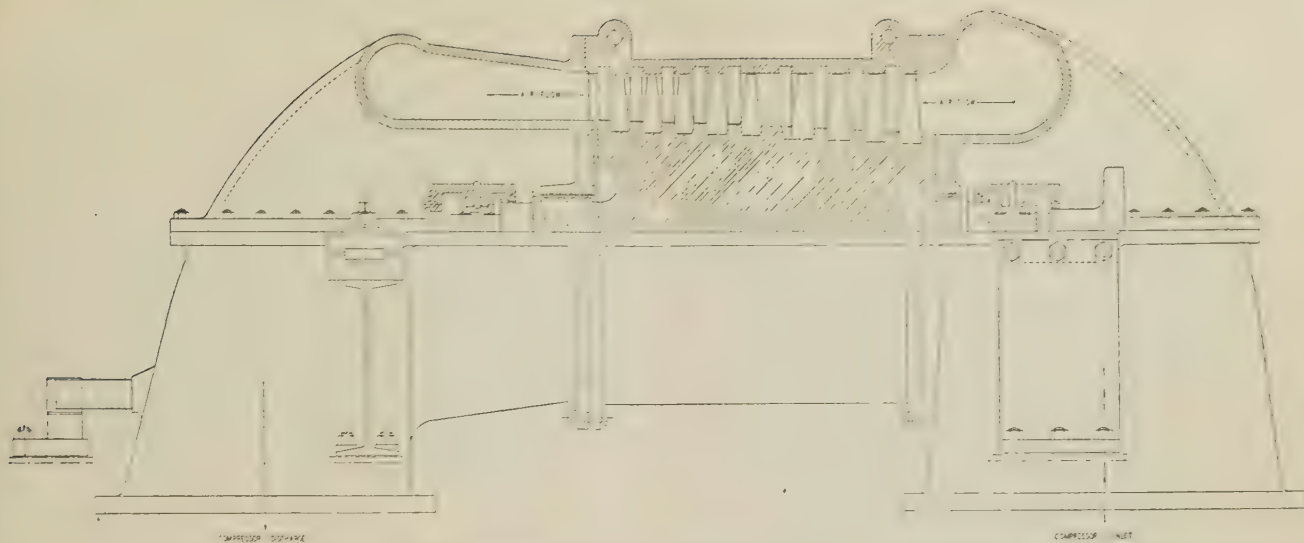


FIG. 4 CROSS-SECTIONAL VIEW OF 9-STAGE AXIAL-FLOW COMPRESSOR

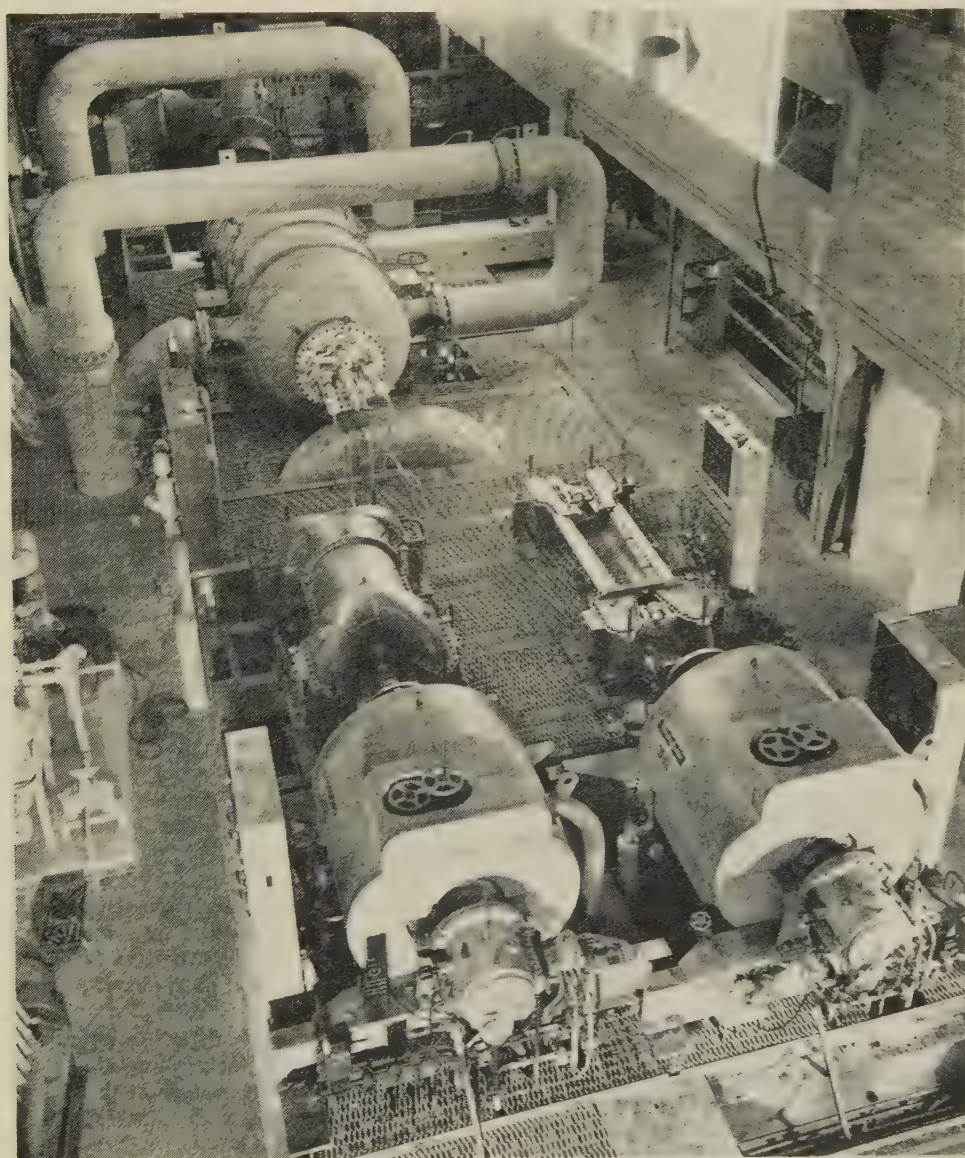


FIG. 5 INSTALLATION VIEW OF AXIAL-FLOW COMPRESSORS
(2:1 at left arranged to supercharge 4:1 at right.)

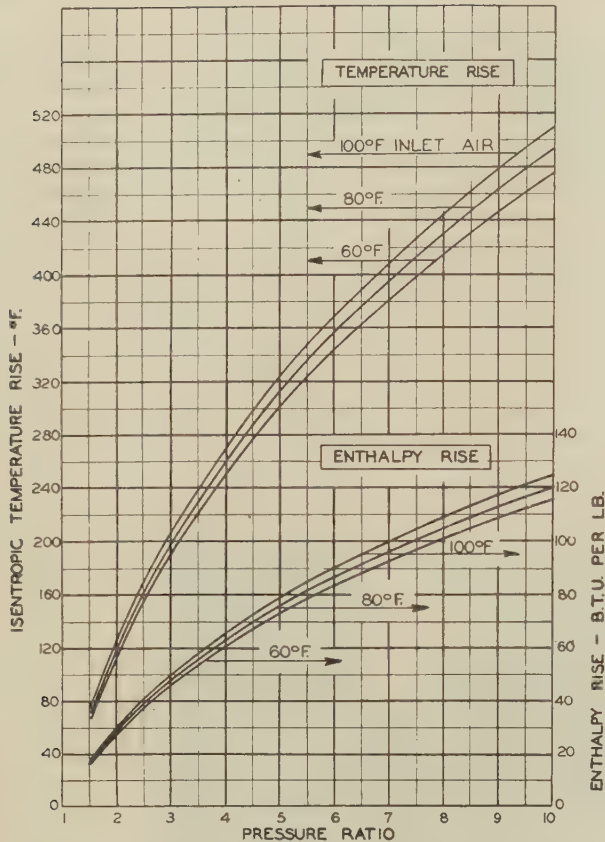


FIG. 6 ISENTROPIC TEMPERATURE AND ENTHALPY RISE FOR AIR (Data from the "Thermodynamic Properties of Air," by J. H. Keenan and Joseph Kaye, 1945.)

$$\eta = \frac{\Delta T_{(\text{isentropic})}}{\Delta T_{(\text{actual})}} = \frac{T_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right]}{T_2 - T_1} \dots \dots [1]$$

In actual applications the compressor may be subjected to operation over a wide range of inlet pressure and temperature conditions, which will affect its performance materially. The turbojet compressor, for instance, must operate at an inlet-temperature range from that existing on the ground to -67°F at 40,000 ft elevation, and the pressure range from atmospheric to 2.7 psia. In order to correlate the compressor performance under variable inlet conditions, the following expressions are used for variables involved in the compressor characteristic curves

$$\text{Mass flow: } W\sqrt{\theta/\delta} \dots \dots \dots [2]$$

$$\text{Volumetric flow: } Q/\sqrt{\theta} \dots \dots \dots [3]$$

$$\text{Speed: } N/\sqrt{\theta} \dots \dots \dots [4]$$

$$\text{Power: } HP/\delta\sqrt{\theta} \dots \dots \dots [5]$$

where θ is ratio of the temperature to an arbitrarily selected standard temperature and δ is the ratio of the inlet pressure to the selected standard pressure. The standard temperature and pressure may be those of the design conditions or any others, such as, standard sea-level conditions of 59°F and 29.92 in. Hg abs adopted by the National Advisory Committee for Aeronautics. The foregoing expressions are derived from the laws of

similitude, and it can be proved that these relations are correct over a wide range of operating conditions, if variations in the value of the gas constant and Reynolds number are neglected.

The compressor operating range at various efficiency levels may be conveniently represented by contour lines of equal efficiencies on the pressure ratio - flow plot, as shown in Fig. 3. Axial-flow compressors of a moderate or high pressure ratio exhibit steep pressure-flow curves. This may impose some limitations as to the operating range for a constant-speed application. In gas-turbine applications, where the temperature and speed control rather than flow control is used, the operating range at any given speed is very narrow and the axial-flow compressor is well adapted. In Fig. 7 the operating range of the 2000-hp gas-turbine generator unit³ for the temperature limits between 1000 and 1300 F is represented by two system pressure ratio - flow lines superimposed on the characteristic curves of its own compressor.

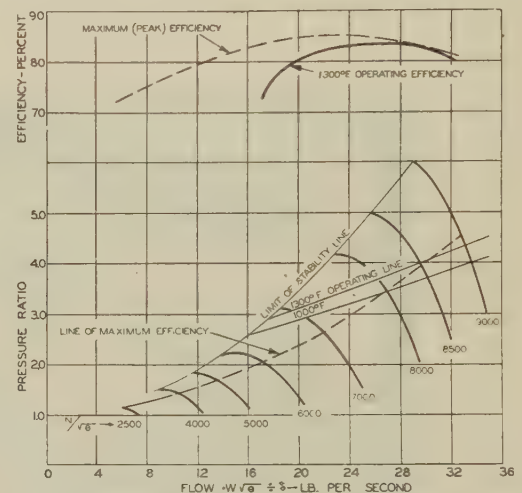


FIG. 7 CHARACTERISTIC CURVES OF AXIAL-FLOW COMPRESSOR APPLIED TO GAS-TURBINE PLANT

Tests of this unit up to date indicate that this 20-stage axial-flow compressor develops the maximum efficiency of about 86 per cent at 7200 rpm. The compressor efficiency at any other speed above or below 7200 rpm is somewhat lower. The reduction in the compressor efficiency at the lower speeds is due to operation of the individual stages off the design velocity ratio, while at the higher speeds the effect of a high Mach number is responsible for the decline of the compressor efficiency.

The operating pressure ratio-flow lines of a variable-speed gas-turbine power-plant system do not follow the line of the compressor maximum efficiency but usually cross it at about the design speed, Fig. 7, and approach the "limit of stability line" at somewhat lower speed. The limit of stability line sometimes referred to as "surge line," or "pumping limit" is formed by the points on the pressure ratio-flow curves at which the compressor ceases to develop a steady discharge pressure. Surges in discharge pressure, accompanied by a large increase in noise and mechanical vibration characteristic of operation beyond this point, sometimes prohibit operation of compressor at the higher speeds.

Stall and Limit of Stability. An airfoil subjected to flow at some small angle of attack will develop a lift force. This lift force will increase with increase of the angle of attack until at some critical angle the maximum lift force is attained. With

³ "A 2000-Horsepower Gas-Turbine Generator Set," by Thomas J. Putz. Presented at the ASME Annual Meeting, New York, N. Y., December 2-6, 1946.

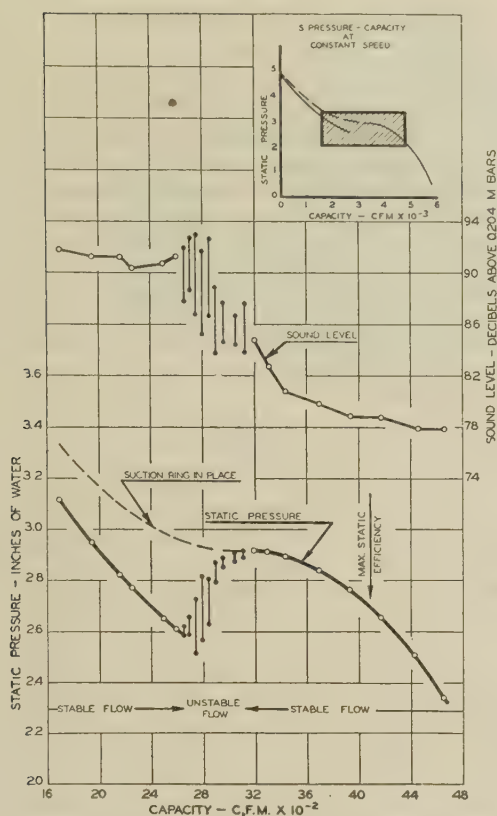


FIG. 8 PRESSURE-CAPACITY CURVE OF AXIAL-FLOW STAGE OPERATING AT LOW SPEED THROUGH RANGE OF INSTABILITY; AND EFFECT OF INSTABILITY ON PRESSURE AND SOUND PULSATIONS

the flow at angles of attack above this critical value, detachment of flow from the surface of the air foil commences at the trailing edge of the convex side with a resultant decrease in the lift force. The airfoil then is said to begin to "stall." With further increase of the angle of attack the flow separation progresses toward the leading edge with a larger decrease in the lift force.

The pressure rise across an axial-flow compressor stage has a definite relation to the lift force of the blade section used (Equation [6]) and varies with the angle of attack. Referring to a pressure-capacity (flow) curve of a compressor stage at a constant speed, Fig. 8, the large capacity and therefore high relative velocity correspond to the small angles of attack and small pressure rise. With the decrease in flow (relative velocity), the angle of attack increases with a resultant higher pressure rise. At the capacity some 20 per cent below that corresponding to the maximum efficiency of the stage, the flow and angle of attack reach their critical value, and a further decrease in capacity results in flow separation from the convex surface of the blade and a drop in pressure.

In a diffusing flow passage formed by two adjacent blades of an axial-flow compressor stage the detachment of flow from the blade surface results in a reversal of flow due to negative pressure gradient. This separation also destroys locally the radial equilibrium in the stage and establishes radial flow due to the centrifugal force.

Compressors usually operate in a system in which the flow is some function of pressure. The flow in such a system establishes the pressure against which the compressor must operate. During the operation at the critical angle of attack, the drop in pressure due to flow separation is accompanied by a reduction of delivery to the system as a result of radial and backflow in the compressor itself. This momentarily re-establishes flow equilibrium which is

destroyed again as the system permits the original flow and pressure conditions. This cycle is repeated with a frequency depending upon the character of the system of which the compressor is a part.

Laboratory investigations of the flow in some axial-flow compressor stages operating beyond the stability limit indicate that, after the flow separation has progressed to cover almost the entire blade surface, a strong reverse flow at the outer periphery of the stage is established. Under these conditions the pressure rise due to the centrifugal force is usually sufficient to balance the system pressure at the levels above the maximum attained by the stage at the limit of stability, and the operation becomes stable and free of pressure pulsation again.

The apparent stabilizing effect of a steady reverse flow during the operation beyond the stability limit led the author's company to the development of a special suction ring, Fig. 9. This suction ring when applied on the suction side of an axial-flow stage

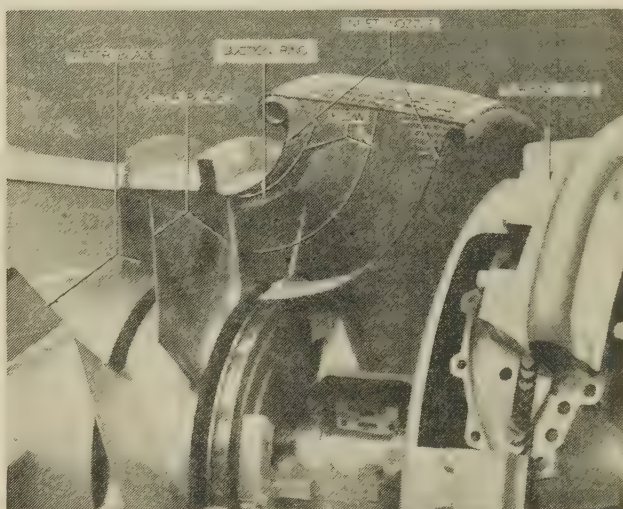


FIG. 9 ARRANGEMENT OF SUCTION RING INSTALLED TO STABILIZE OPERATION OF AXIAL-FLOW BLOWER

prevents recirculation of the reversed flow air across the rotating row, eliminates pulsation of the pressure, and is responsible for producing a slowly rising pressure-capacity curve throughout the zone of instability, as shown by a dash line in Fig. 8. The use of such a suction ring made possible the installation of a very large number of single-stage axial-flow blowers for forced-draft application on ships of the U. S. Navy, under a rigid specification in respect to the stability of operation over the entire range from the shutoff to the maximum delivery. The suction ring was applied successfully to two- and three-stage axial-flow blowers, designed for static pressures in excess of 100 in. of water. The application of a modified form of such suction ring to a multistage axial-flow compressor is being investigated. Fig. 10 shows the characteristic curve at a constant speed obtained from the tests of a three-stage axial-flow blower.

The inherent instability of an axial-flow compressor during the operation beyond the limit of stability line does not present a major problem when such compressor is used in a gas-turbine power plant. Usually it is possible to arrange a by-pass control on a compressor or a temperature control in the gas turbine to keep operating conditions within the limits of stable operation. Furthermore, compressor proportions can be selected so that the operating system line always lies under the limit of stability line or crosses it at low speeds where the pressure surges are mild and safe.

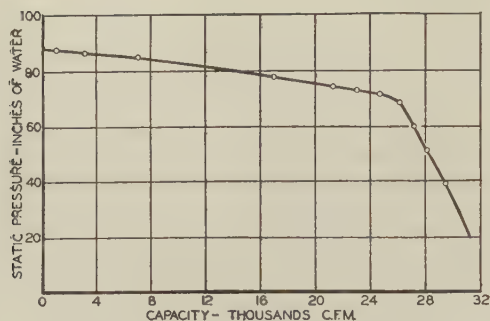


FIG. 10 PRESSURE-CAPACITY CURVE OF 3-STAGE AXIAL-FLOW BLOWER, EQUIPPED WITH SUCTION RING

TYPES OF AXIAL-FLOW COMPRESSORS

In the design of an axial-flow compressor a great many different arrangements of blades and velocity diagrams may be employed effectively. Axial-flow compressors which have been used successfully here and in Europe may be classified broadly into two types; symmetric or constant reaction, and nonsymmetric or vortex. The nonsymmetric-stage blading may be arranged for air to enter the rotating row either axially or with a swirl in the direction opposite to rotation. The orientation of rotating and stationary-row blades, velocity vectors, and pressure rise characteristics for symmetric and two types of nonsymmetric stages are shown in Fig. 11. For the purpose of illustration of their salient characteristics, the same blade form for the rotating rows and equal values of axial velocities are assumed for each type. Under these conditions the blade section

of the rotating row will produce the same pressure rise for all three types, and the stage performance can be compared on the basis of the blade speed and the reaction of the stationary blades.

Symmetric-Stage Compressor. In a symmetric stage, often referred to as constant reaction, air enters the rotating row with an initial swirl in the direction of rotation. The orientation of the blades and their sections in the rotating and stationary rows on the same diameter are identical but mirror images. Fig. 11 shows that the vectors of absolute air velocities entering, C_1 , and leaving, C_4 , the rotating row are symmetrical with the relative velocities C_2 and C_3 , indicating that the pressure rise in the stationary row is equal to that obtained in the rotating row. The equal pressure rise in the rotating and stationary rows leads to still another name for the symmetric-stage compressor, namely, 50 per cent reaction.

In the symmetric type of compressor, the initial swirl in the direction of rotation permits the use of high air velocities and blade speeds without producing high velocities relative to the rotating blades which may be detrimental to compressor efficiency as a result of Mach number effect. The aerodynamic design of this type of compressor requires a higher axial velocity at the base of the blade, which tends to produce an equal relative velocity or Mach number along the blade height. Blade-tip speeds exceeding the velocity of sound, and axial air velocities up to 700 fps have been employed successfully by designers of the symmetric type of compressor. High air velocities and blade-tip speeds of symmetric compressors result in small over-all dimensions and high rotative speeds, while a large pressure rise in stationary blades keeps at a minimum the number of stages required for a given pressure ratio. These features make the symmetric compressor highly suitable for aviation gas-turbine

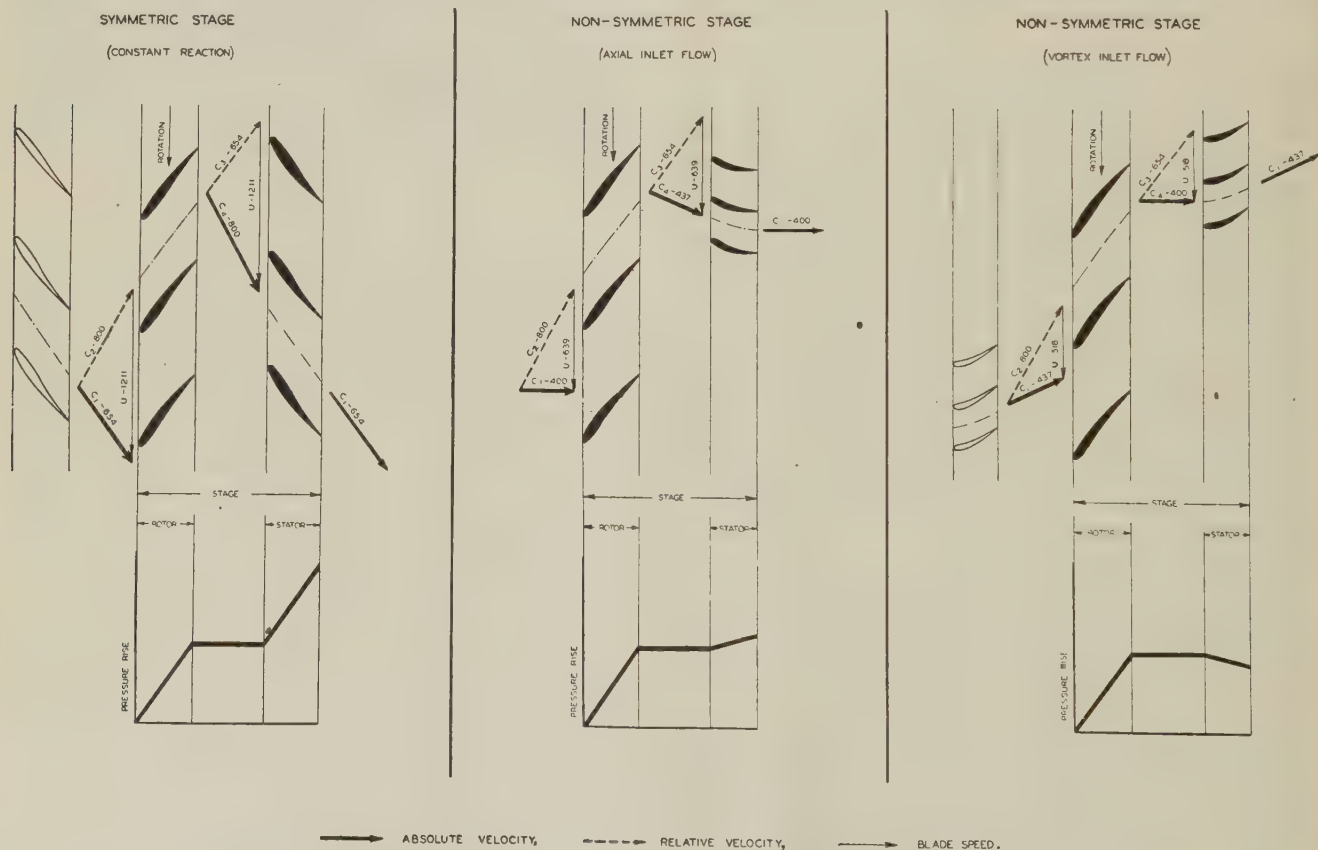


FIG. 11 BLADING ARRANGEMENT USED ON SYMMETRIC AND NONSYMMETRIC TYPES OF COMPRESSORS, SHOWING PRESSURE RISE ACROSS A STAGE, AND VELOCITY VECTOR DIAGRAMS

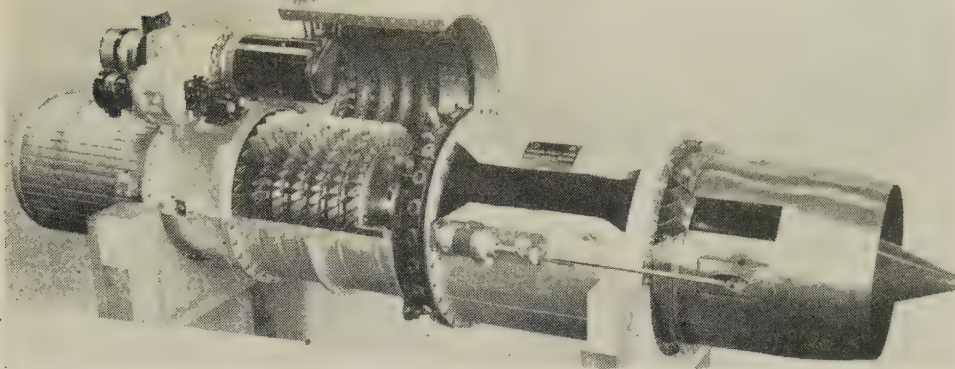


FIG. 12 A 19-IN. TURBOJET UNIT WITH COMPRESSOR COVER REMOVED TO SHOW DETAILS OF SYMMETRIC-TYPE COMPRESSOR

applications. The 19-in. turbojet built by the author's company and used by the U. S. Navy to power the McDonnell FD1 "Phantom" is a good example of such an application, Fig. 12. In this 19-in. turbojet, a six-stage 15 $\frac{3}{4}$ -in-tip-diam symmetric-type compressor delivers 25,000 cfm of air at a pressure ratio of 3.5:1 when operating at a speed of 18,000 rpm.

In the field of land transportation and in marine applications where high efficiency and reliability throughout a long operating life rather than small weight and frontal area are of the utmost importance, the symmetric compressor does not occupy the leading position. High axial air velocities result in high leaving losses and unfavorable blade aspect ratio. High pressure rise across the stationary rows requires an intricate sealing arrangement to reduce leakage and to eliminate the flow disturbance at the base of rotating blades. These features and, perhaps, the extreme complexity of flow pattern of the symmetric type of compressor are responsible for selection of the vortex or nonsymmetric type of compressor design for many gas-turbine power plants for land and marine applications.

Nonsymmetric-Stage Compressor. In a nonsymmetric stage, the vectors representing the absolute air velocities entering and leaving the stationary row are not symmetrical with those representing the relative velocities C_2 and C_3 in Fig. 11. The blade sections in the rotating row are arranged to produce a constant axial velocity and constant work at all diameters of the stage. As a result of this the absolute tangential component (swirl) is made to vary inversely as the diameter, in a manner similar to the so-called "free-vortex" motion of fluid in an unrestricted space. The conditions of the free-vortex flow may be found in the following types of nonsymmetric stages:

- 1 Axial inlet stage.
- 2 Vortex inlet flow stage.

In (1) the air enters the rotating stage axially and the free-vortex flow is established after the rotating row. In (2) the free-vortex flow is established by the stationary row by imparting a tangential component (inversely as the diameter) in direction opposite to rotation. In this case the air leaves the rotating row axially.

Axial Inlet Stage. In this type of nonsymmetric compressor the stationary blades are designed to give a pure axial flow at the exit, and the sections of the rotating blades are oriented to produce a swirl the magnitude of which varies inversely as the diameter. The pressure rise in the stationary row, obtained through removal of the swirling component of the air velocity is very small, and this compressor stage is at times referred to as a "ten per cent reaction," which implies that only 10 per cent of the stage pressure rise is obtained in the stationary row. Small

pressure rise across the stationary rows permits the use of constant-section stationary blades, as well as the omission of the inter-stage seals with but a small sacrifice in the over-all compressor efficiency.

Axial velocities up to 400 fps and blade speeds up to 750 fps are used with this type of compressor. These relatively low axial velocities and blade speeds are conducive to the lower leaving losses, good blade aspect ratio, and moderate rotative speeds suitable for direct drive by a gas turbine. The experimental 2000-hp gas-turbine generator unit of the author's company, Figs. 13 and 14, is served by a 20-stage nonsymmetric compressor of this type. Simplicity of the flow pattern and design make this type of stage very attractive for use on single-stage applications, such as blowers and ventilating fans. The recent tests of the 9-stage compressor of this type indicated efficiencies up to 88 per cent.

Vortex Inlet Stage. In this type of nonsymmetric compressor the blades in the stationary rows are arranged to produce a swirl in a direction opposite to rotation. The blades in the rotating rows are designed to remove this swirl and re-establish axial velocity at the entrance to the stationary row of the succeeding stage. Inspection of the velocity triangles in Fig. 11 shows that high relative velocities may be obtained even with low axial velocities and blade speeds. Adverse effects of high Mach number limit the velocities and blade speeds of this type of compressor stage to very low values. Axial velocities as low as 200 fps, and blade speeds below 500 fps are common.

Low axial velocities and blade speeds result in large dimensions and low rotating speeds. These features make this type of compressor extremely suitable for the low volumetric capacities which are encountered in closed-cycle gas-turbine applications where the air enters the compressor at some elevated pressure. The Escher Wyss-AK closed-cycle turbines, described by Keller,⁴ are served by the vortex-inlet-stage compressors.

Negative reaction or pressure drop in the stationary rows, and limitations as to the velocities result in a larger number of stages as compared with the other types of axial-flow compressor. The large number of stages, however, does not reflect too unfavorably on the axial length of the compressor as the low blade speeds and loading permit the use of smaller blade chords. A favorable blade aspect ratio, low air velocity at the discharge from the last stage, and relatively small effect of radial blade clearances contribute to the high efficiencies obtainable on this type of nonsymmetric compressor.

⁴ "The Escher Wyss-AK Closed-Cycle Turbine, Its Actual Development and Future Prospects," by Curt Keller, Trans. ASME, vol. 68, 1946, pp. 791-812.

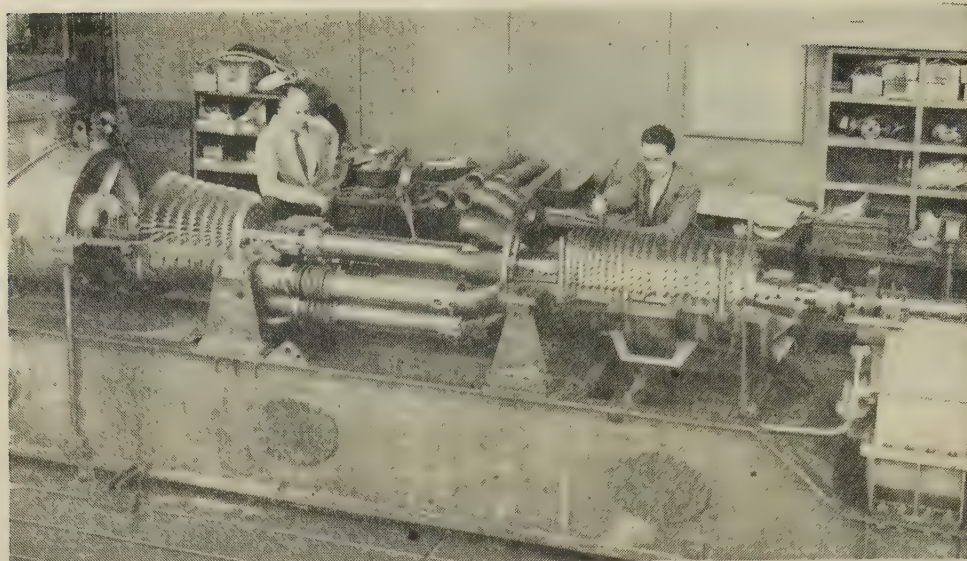


FIG. 13 EXPERIMENTAL 2000-HP GAS-TURBINE UNIT

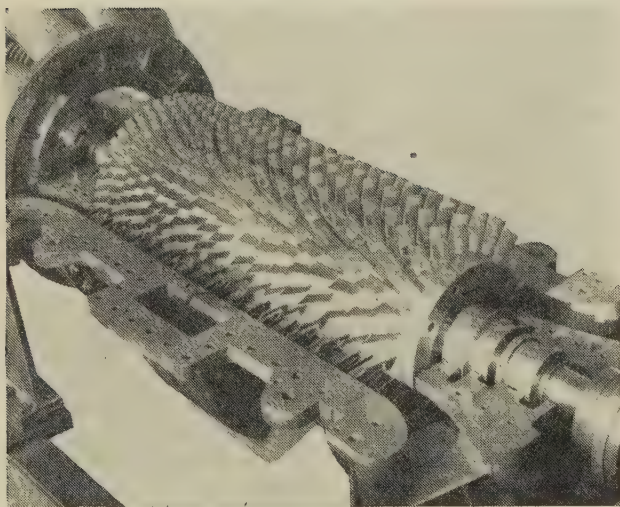


FIG. 14 NONSYMMETRIC 20-STAGE COMPRESSOR FOR 2000-HP EXPERIMENTAL GAS-TURBINE UNIT

FACTORS AFFECTING PERFORMANCE OF AXIAL-FLOW COMPRESSOR

The volumetric flow, pressure-generating capacity, and the efficiency of an axial-flow compressor with any chosen type of velocity diagram, depend upon many variables. Some of these variables affect the compressor performance because of the geometric configuration of blades in the stages, while the others do so because of the dynamics of the fluid flow established by the arrangement of the blades and their speeds. Among these variables the most important are the following: (1) Velocity ratio; (2) blade spacing; (3) hub-tip diameter ratio and blade aspect ratio; (4) reheat; (5) Mach number; (6) Reynolds number.

Velocity Ratio. It is well known that the relation between the fluid velocity and blade speed of a steam or hydraulic turbine determines the basic efficiency of the machine. In a similar way the ratio of the air velocity to blade speed has a pronounced effect on the efficiency of the axial-flow compressor.

In the design of an axial-flow compressor for a given flow,

pressure ratio, and speed, a favorable velocity ratio, conducive to the maximum efficiency, is usually selected for all stages by an arbitrary assignment of the axial velocities and blade speeds. With the selected axial air velocities, the flow area in all stages is established from the pressure and temperature conditions at each stage. During operation at reduced speeds the compressor discharge pressure falls very rapidly with the speed, and the volumetric flow through the compressor stages does not decrease as rapidly. Since the flow area is proportioned for the design speed and compression ratio, this results in a progressive increase in the velocity ratio from the first to the last compressor stage with a resultant drop in the compressor over-all efficiency as shown in Figs. 3 and 7. The operation of the individual compressor stages when off their design velocity ratios results also in the flatter pressure-flow characteristic curves at the reduced speeds.

At the design speed the axial velocities and amount of turning corresponding to the assigned velocity diagrams at various diameters of the compressor stage are obtained through the orientation of the blade sections at the proper angle with the axis of rotation (stagger angle), and the selection of the correct blade-camber angle (illustrated in nomenclature). This usually leads to twisted blades of a variable camber for both rotating and stationary rows, while the blade-stress considerations dictate blade sections tapered in area from the base to tip.

Pitch-Chord Ratio. In an axial-flow compressor the blade section, stagger angle, and pitch-chord ratio describe completely the geometry of the air-flow path in the rotating and stationary rows. The stagger angle γ (see nomenclature) or orientation of the blade in respect to the axis of rotation, determines the magnitude of the axial air velocity. The camber angle θ of the blade section sets the amount of turning which this section will produce. In an axial-flow compressor the amount of turning which the given camber can produce depends to a large extent upon the spacing between two adjacent blades of a given chord length. The designers of steam turbines and axial-flow compressors usually express the blade spacing in terms of ratio of the pitch of blades to the blade chord at the same diameter or pitch-chord ratio.

It can be shown that the amount of turning, lift coefficient, and pitch-chord ratio of the compressor blades are related as follows

$$\frac{S}{L} = C_L \frac{\sqrt{\left(\frac{\tan \beta + \tan \alpha}{2}\right)^2 + 1}}{2(\tan \beta - \tan \alpha)} \dots \dots \dots [6]$$

The effect of the pitch-chord ratio (S/L), on the performance of the blade section of a given camber has been the subject of extensive experimental investigations here and abroad. The cascade-flow tests of various blade arrangements, as well as the actual tests of rotating compressor stages, indicate that the amount of turning affected by the blade grid of a given blade section is always smaller than the camber angle of the blade section.

The amount of this deviation in the turning angle is a function of the pitch-chord ratio and can be approximated from the following empirical formula

$$\beta - \alpha = \theta(1 - 0.25\sqrt{S/L}) \dots \dots \dots [7]$$

This correction factor, as suggested by A. R. Howell⁵ in a somewhat different form, has been applied by the author for the selection of the blade camber for the desired turning, with satisfactory results.

Hub-Tip Diameter and Aspect Ratios. The flow path of an axial-flow compressor is designed according to the "continuity equation" with the flow area gradually decreasing toward the discharge end, corresponding to the assigned velocity, actual pressure and temperature state at each stage. This results in a gradual decrease in the blade height and an increase in the ratio of the compressor rotor hub to the blade-tip diameters. The decrease in the blade height is usually accompanied by a decrease of the aspect ratio or ratio of blade height to the chord length.

The tip and base of the compressor blade operate in boundary layers formed at the bounding walls of the flow passage and the performance of these portions of the blade suffers to a considerable extent, owing to the prevailing low velocities. This inactive portion of the blade becomes larger in proportion to the total height of the blade as the hub-tip diameter ratio increases toward the discharge end of the compressor, with a resultant decline in efficiency.

Tests conducted by many investigators indicate that the efficiency of an axial-flow compressor deteriorates rapidly with the increase of the hub-tip diameter above a value of 0.85. The hub-tip diameter ratios in excess of 0.9 should be avoided unless the flow is sufficiently large to permit the use of relatively long blades.

This limitation as to the maximum hub-tip diameter ratio provides the incentive for the use of smaller hub diameters for the first compressor stages, especially when a large pressure ratio is involved. The use of a small hub diameter usually leads to undesirable blade spacing at the hub, and at times to some mechanical difficulties in design of the blade attachment. These two factors in many cases limit the lower value of the hub-tip diameter ratio to 0.5.

Reheat. In a steam or gas-turbine stage losses reappear in the form of heat energy part of which is available for work in the succeeding stages. The reheat effect results in a better over-all internal efficiency as compared with the individual stage efficiency. The designer of a compressor is in a less fortunate position. The friction losses of a compressor stage produce the effect of preheating, which decreases the over-all compressor efficiency. The effect of reheat on compressor efficiency depends upon the operating pressure ratio and stage efficiency as shown in Fig. 15.

Mach Number. Subacoustic flow of fluids through a passage or around a body completely immersed in that fluid follows a cer-

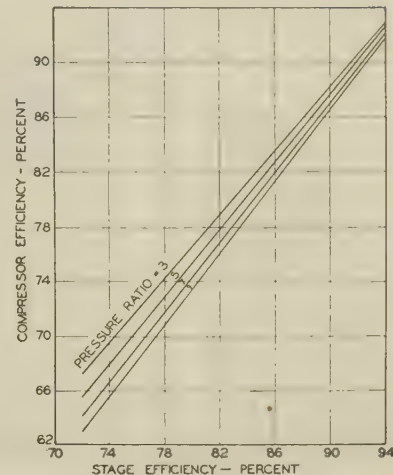


FIG. 15 EFFECT OF AIR PREHEAT IN COMPRESSOR STAGES ON OVER-ALL COMPRESSOR EFFICIENCY

tain pattern depending upon the geometry of the passage and the Reynolds number. At the velocity equal to that of sound shock waves form, and a complete change in flow pattern takes place. In the passage designed for subacoustic flow, such as in an axial-flow-compressor stage, this change of flow pattern is accompanied by a serious loss known as the losses due to the Mach number effect.

When applied to the performance of a compressor, the Mach number is defined as the ratio of the mean air velocity relative to the blade to the local velocity of sound. With this accepted definition it is apparent that the Mach number effect on the compressor stage performance would be observed in the operation at Mach number values below 1.00. Large deviation of the local velocities from the mean flow velocity, due to the flow pattern around an airfoil exerting lift, results in the fact that even in a well-designed compressor stage the Mach number as low as 0.60 may show detrimental effect on the efficiency.

The Mach number effect manifests itself in a pronounced reduction in compressor efficiency, and in reduction of flow coefficient. When operating at high mean velocities corresponding to the Mach number above the critical value, the increase of compressor speed does not produce the expected increase in flow. This results in the steeper pressure ratio - flow curve which may be observed at the higher speeds of the compressor. As the Mach number approaches the value of 1.00, an axial-flow compressor reaches its limiting flow, and the pressure ratio - flow curve becomes the vertical line of the constant flow machine.

Reynolds Number. In evaluating performance of axial-flow compressors of various designs, especially the performance of a low-speed model, another dimensionless criterion of the flow dynamics must be taken into account, namely, Reynolds number. The Reynolds number is well known in fields of hydraulics and aerodynamics as a scale effect. In the physical sense, the Reynolds number is the ratio of the inertia forces to the viscous forces acting on fluid particles. A small Reynolds number, say, 1500, for flow of oil in a pipe indicates viscous flow or a predominant effect of fluid viscosity on the flow pattern, while a high Reynolds number, say, 500,000, of air flow in a compressor indicates turbulent flow and the preponderance of inertia forces.

When applied to the flow through an axial-flow-compressor stage, the Reynolds number is expressed as a ratio of the product of blade chord and mean relative velocity to the kinematic viscosity of fluid, or

$$R_e = \frac{LCm}{\nu} \dots \dots \dots [8]$$

⁵ "Development of the British Gas Turbine Jet Unit," The Institution of Mechanical Engineers, London, England; ASME reprint for distribution in the United States, 1947.

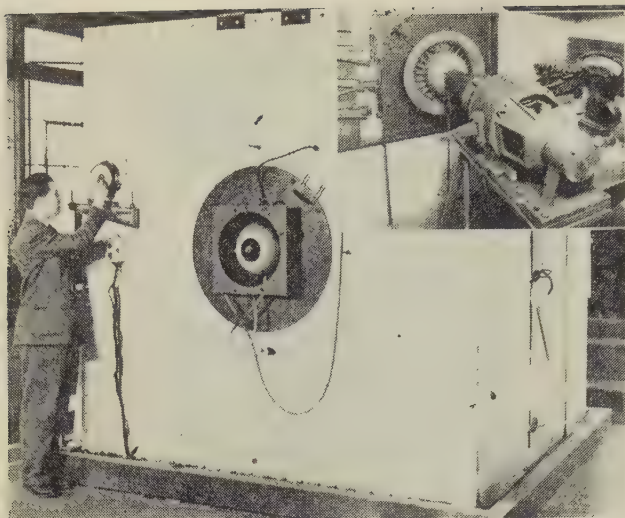


FIG. 16 LOW-SPEED-COMPRESSOR MODEL TEST TUNNEL
(Insert shows arrangement of dynamometer and test gages.)

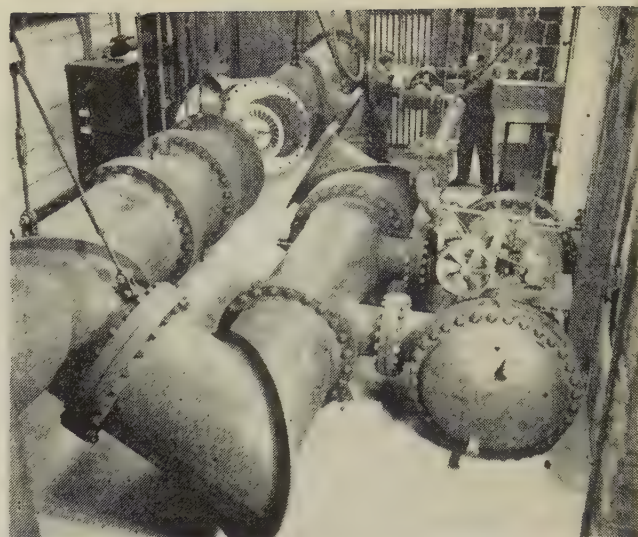


FIG. 18 VARIABLE-DENSITY WIND TUNNEL WITH TEST SECTION
PARTIALLY REMOVED

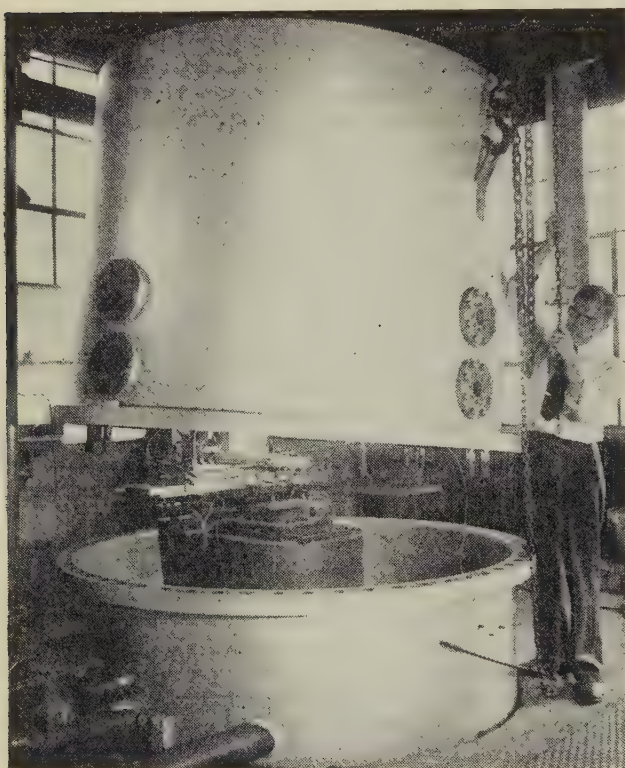


FIG. 17 VARIABLE-DENSITY CASCADE WIND TUNNEL

Axial-flow compressor blades usually operate at Reynolds numbers above 500,000, while a slow-speed stage model may operate at but one tenth of that value. As both high Reynolds numbers and Mach numbers are associated with high relative velocities, the use of variable-density tunnels for separation of these two effects on the performance of airfoil or compressors stage is widely employed. Figs. 16, 17, and 18 show the three types of wind tunnels extensively used by the South Philadelphia Works of the author's company for aerodynamic research.

The isometric diagram, Fig. 19, shows the effect of the Mach and Reynolds numbers on the efficiency of a nonsymmetric type compressor tested in the variable-density tunnel.

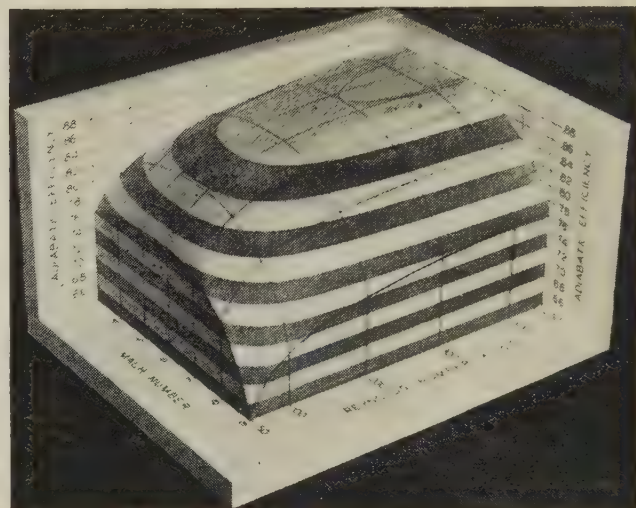


FIG. 19 THREE-DIMENSIONAL DIAGRAM SHOWING EFFECTS OF
MACH AND REYNOLDS NUMBERS ON AXIAL-FLOW COMPRESSOR
EFFICIENCY

CONCLUSION

The gas-turbine power plant is the product of development of the last few years. The development of the aviation gas-turbine plant was conducted at an accelerated rate because of military necessity and the inherent limitations of the reciprocating engine. The aviation gas-turbine plant has already become a reality. In the fields of marine transportation and land applications, the gas-turbine power plant is still in the experimental stage, and its reduction to practice will require the solution of a great many problems by planned research and development.

At the present time, when all gas-turbine power-plant problems have not been fully explored as yet, the axial-flow compressor stands out as a natural choice for its essential component as being highly efficient, light in weight, and well adaptable to the service of system compressor. Tremendous strides have been made in the development of the axial-flow compressor during the last 5 years. However, many problems remain to be solved. The most important one being the design of an efficient compressor which can be manufactured at a low cost, so that the gas-turbine power plant with its compressor may compete with the highly developed steam turbines and reciprocating engines.

Discussion

G. M. DUSINBERRE.⁶ Since there is not yet a widespread familiarity with the vector diagrams of the axial-flow compressors, it may be helpful to offer the following extension of the author's treatment.

The two diagrams for the symmetric stage may be combined as shown in Fig. 20 of this discussion. This gives us the important vector AB , which determines the tangential force on the blading, $175/g = 5.43$ lb, per lb of gas per sec. Combining this with the blade speed gives the work, $175 \times 1211/778g = 8.46$ Btu per lb, and, in connection with the efficiency, gives the stage pressure rise. Angles ACB' and BDA' determine the blade shapes (incidence and deviation being here neglected).

In the author's analysis the moving blade shape and axial velocity have been left unchanged. Hence the vector triangle ABC is unchanged and all the author's diagrams can be combined as in Fig. 21 of this discussion. Here we can readily visualize the decreased work per stage (requiring more stages for the same pressure rise) and also the altered shapes required for the fixed blades.

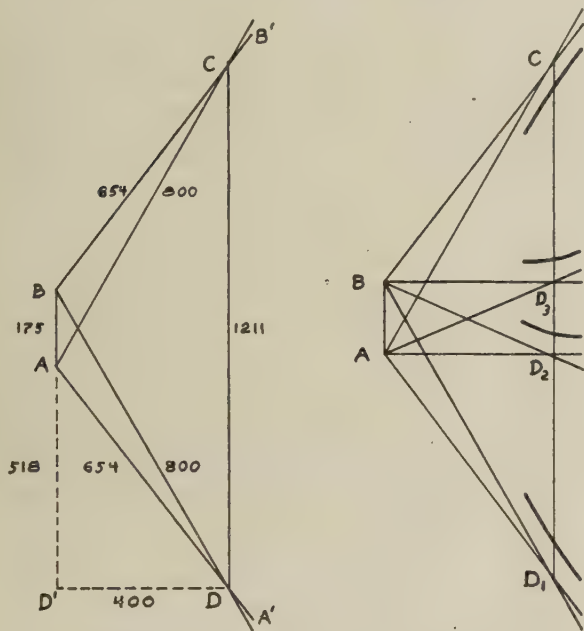


FIG. 21 THREE DESIGNS WITH
COMMON DEFLECTION IN MOVING
BLADES

It is evident that we have to have vortex flow at some point in any case. The question whether we have a "free" vortex or not depends on conditions along the length of the blades.

We might expect to avoid radial flow by providing for the same rise of total pressure at all points along the blade length. With constant efficiency, this means equal work at all points. Since the blade speed varies directly as the radius, the vector AB should then vary inversely as the radius for equal work. If the present diagrams are intended to represent conditions at mean blade radius, say, $r = 0.9$, then at $r = 1.0$, $AB = 158$, and at $r = 0.8$, $AB = 197$. This condition will call for "twist" in the blading.

But if we wish to provide for free vortexes, we must meet the conditions noted by the author. This can be done by varying the vector AD' in the same proportion as the vector AB , in other words, "sliding" the vector AB to an appropriate position relative

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to CD . Thus we are not necessarily limited to the second two diagrams of Fig. 11 of the paper, as a basis for free-vortex design. The first diagram might also be used, as shown in Fig. 22 of this discussion. The construction is simpler in the other cases because AD' or BD' is zero.

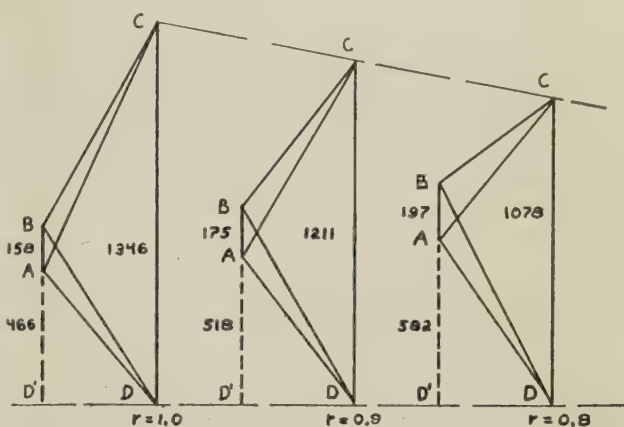


FIG. 22 FREE-VORTEX DESIGN SYMMETRICAL AT MEAN RADIUS

One general principle can be seen from any diagram of this type: Other things equal, the greater the axial velocity, the less will be the deflections required.

Everyone interested in gas turbines will have arrived at some scheme similar to the foregoing; the writer's only excuse for offering it is that the presentation has been found useful in classroom and may be helpful to engineers who only recently have become interested in this field.

As a minor correction, the blade speed in the author's second diagram, Fig. 11, should be 693 fps, not 639.

A. J. STEPANOFF.⁷ In discussing performance of the axial-flow blower shown in Figs. 8 and 9, the author attaches too much importance to the ring baffle in the blower inlet. There are numerous blowers on the market, nationally advertised, which obtain a constantly rising head-capacity characteristic without the use of such a baffle. All these (including the author's in Fig. 9) approach hydraulically the design of axial-flow pumps, i.e., have higher impeller discharge and stagger angles, fewer and longer impeller vanes, greater axial length per stage. It might be mentioned that early centrifugal and axial-flow pumps (Wood patent) had unstable regions in their characteristics. However, modern pumps are free from these defects.

The blade-design procedure outlined by the author is based on "streamline" or "blade-element" theory. A free-vortex velocity distribution is assumed for velocity diagrams. It is a general belief that only a free vortex is stable radially. The writer has demonstrated⁸ that a free vortex is only one of many possible types of vortexes, depending upon the angular velocity distribution along the radius, all of which are stable radially. The writer would like to call attention to the "forced vortex" as a pattern of flow for centrifugal and axial-flow pumping machinery (water or air). The definition and properties of a forced vortex are as follows:

1 The absolute angular velocity of flow through the machine is constant along the radius. This applies to guide vanes, impeller, and stationary vanes.

⁷ Development Engineer, Ingersoll-Rand Company, Phillipsburg, N. J. Mem. ASME.

⁸ "Centrifugal and Axial Flow Pumps," by A. J. Stepanoff, John Wiley & Sons, New York, N. Y., 1948, p. 14.

2 The same pattern of flow (forced vortex) prevails at all rates of flow from zero capacity to zero head points.

3 For axial-flow machines, the head produced is greater at the impeller periphery than at the impeller hub. This is equalized when the tangential component is taken out of the flow by the stationary vanes by "conduction" (without "crossflows"), a term borrowed from heat transmission. Both pressure energy and heat energy are due to molecular kinetic energy, and are interchangeable, as is well known in thermodynamics of gases.⁹

4 When applied to axial-flow machines, the forced-vortex pattern of flow produces the same "dimensionless head" along the radius, or

$$\psi = H \frac{u_2}{2g} = \text{const}$$

where H is the total head and u is the peripheral velocity.

5 The forced vortex results in identical theoretical treatment and impeller design procedure for straight centrifugal, mixed-flow, and axial-flow machines.

Based upon the forced-vortex reasoning, the writer has constructed a diagram,¹⁰ which incorporates in it performance characteristics, controlling design elements, actual and theoretical velocity triangles for all pumping machines, i.e., centrifugal, mixed-flow, and axial-flow.

By a comparison, the free-vortex pattern of flow is maintained at one point on the head-capacity curve only. At partial capacities, the actual total head of the machine is frequently higher than u^2/g (maximum possible theoretical head) at the hub of the impeller, which has no sense. Expressions of dissatisfaction of the free-vortex theory and deviations from it ("half-vortex") have appeared repeatedly in recent writings.¹¹ There are some scattered bits of information on forced-vortex pattern of flow in literature. A definite statement by Hayne Constant is found in author's reference (5).¹²

The method of the impeller-vane layout, such as prescribed by the forced vortex, was used many years ago. Schmidt¹³ used it for a blower design prior to 1928. His theoretical reasoning was entirely different, and the writer confesses that he has been unable to understand it to the present day.

AUTHOR'S CLOSURE

The lack of a widespread familiarity with vector diagrams in general led the author to the presentation of the vector diagrams for the axial-flow compressors in the form shown on Fig. 11, with the emphasis placed on the blading arrangement rather than on the velocity vectors. Mr. Dusingberre's treatment of the vector diagrams is a valuable contribution to the understanding of the flow dynamics of an axial-flow compressor. The combined diagrams, Figs. 20 and 21, bring out important vectors describing the work and the pressure rise across a compressor stage.

⁹ "Centrifugal and Axial Flow Pumps," by A. J. Stepanoff, John Wiley & Sons, New York, N. Y., 1948, p. 11.

¹⁰ Ibid, p. 185.

¹¹ Author's reference (5), pp. 421 and 456. See also "Minutes of Axial Flow Compressor Meeting of June 26 on German Articles," by C. W. Smith, Bureau of Ships Code 445A, Navy Department, 1946, p. 40.

¹² Ibid., p. 421.

¹³ "Some Screw Propeller Experiments," *Journal of the American Society of Naval Engineers*, vol. 40, no. 1, February, 1928.

Mr. Dusingberre's diagram of a free vortex design, symmetric at mean radius, Fig. 22, is one of the many that are at disposal of the axial-flow compressor designers. This design permits a high pressure rise per stage with the added benefit of the free vortex flow pattern.

The author is indebted to Mr. Dusingberre for calling attention to the numerical error in the vector diagram of Fig. 11.

The author does not agree with Dr. Stepanoff's statement that modern pumps and blowers exhibit a constant rising head-capacity curve and are free of unstable regions in their characteristics. It is true that there are some low-pressure fans and blowers on the market that may be operated free of instability throughout the entire operating range. However, such blowers are usually designed for a low-pressure rise per stage, and the application of their features obviously cannot be justified in a multistage compressor. The designer of modern centrifugal or axial-flow-type compressors is still facing a difficult problem of flow stability in a certain operating range. The same holds true in the case of axial-flow pumps. In this connection the author could not fail to note that the recently installed 24 in. axial-flow pumps, the construction and tests of which are credited to Dr. Stepanoff,¹⁴ exhibit unstable head-capacity and hp-capacity characteristics at all runner vane settings. The sharp break and the reversal of curvature in the characteristic curves of these pumps point definitely to blade stall and unstable flow conditions, which the author believes may be eliminated through the application of the "suction ring," such as shown on Fig. 9.

Dr. Stepanoff is to be complimented on his untiring efforts in advocating the hydraulic treatment of all pumps as one continuous series, irrespective of their runner designs, with the specific speed,¹⁵ as an index number indicative of the type. This hydraulic family includes the low specific-speed centrifugal pumps, medium speed mixed-flow and the high specific-speed axial-flow or propeller pumps. The benefit derived from such treatment is generally recognized now by both pump designers and pump users. It is believed that the similar treatment of compressors will be a welcome addition to the technical literature that will lead to a further advancement of the art.

The author agrees that the forced vortex flow pattern advocated by Dr. Stepanoff may be successfully used, as well as many other patterns, in designing of an axial-flow compressor. The author has no reason, however, to express dissatisfaction with the free-vortex blade design and theory, especially in view of the excellent performance reported by many independent investigators of this type of compressor. The test results of 9-stage axial-flow compressor, Fig. 3, indicated a remarkably close agreement with the calculated performance. The fact that the theory based on the free-vortex flow pattern does not render the performance calculations possible beyond the "stall" region or near the shutoff is of secondary importance.

The author regrets that at the time of this writing Dr. Stepanoff's "Centrifugal and Axial-Flow Pumps," to which the numerous references in discussion were made, was not out of press and the author has not had the opportunity to examine what undoubtedly will be an excellent treatise on pumps.

¹⁴ "Adjustable Vanes Match Pumps," *Power*, March, 1948, pp. 92 and 93.

¹⁵ "Centrifugal-Pump Performance as a Function of Specific Speed," by A. J. Stepanoff, *Trans. ASME*, vol. 65, 1943, pp. 629-647.

Ignition and Flame Stabilization in Gases¹

By BERNARD LEWIS² AND GUENTHER VON ELBE,³ PITTSBURGH, PA.

The self-ignition of a volume of explosive gas mixture is understood to be the transition from slow to very rapid chemical reaction with approximately adiabatic heat evolution. Two known processes bring this about. They are distinguished as thermal and branched-chain explosion. In the former, the chemical reaction becomes self-accelerating owing to the temperature rise. In the latter, the self-acceleration is caused by the formation of chemically active particles at a rate that exceeds the rate of destruction of these particles. A peculiar process of the latter type plays an important contributory part in engine knock. Local ignition, as by a spark, results in the formation of a combustion wave which propagates through the explosive mixture in a manner analogous to Huyghen's principle for the propagation of a light wave. Small sparks do not produce ignition. The minimum ignition requirement of electric capacitance sparks in mixtures of methane or other hydrocarbons with oxygen and nitrogen is found to be only the spark energy. This is theoretically explained, and the minimum spark-ignition energy is shown to be a function of the burning velocity, the width of the combustion wave, and other variables of the gas mixture in agreement with experimental data. A combustion wave in a stream of explosive gas mixture becomes stationary, i.e., the flame becomes stable, when the burning velocity is equal to the gas velocity somewhere in the wave and is nowhere larger than the gas velocity. The mechanism is described by which the condition of equality of burning velocity and gas velocity is realized at the rim of a burner tube. It is shown that the limits for flash back and blowoff correspond to critical values of the gas-velocity gradient at the stream boundary. This applies equally to the outer boundary and any inner boundary formed by solid objects in the stream.

AN explosive mixture of fuel and oxygen may be ignited essentially in two ways, as follows:

1 The mixture is passed into a vessel, or through a tube, and subjected in bulk to high temperature or pressure. Under such conditions ignition occurs almost simultaneously in every part of the vessel after a time lag ranging from less than a millisecond to hours, according to circumstances. This type of ignition may, for brevity's sake, be termed "self-ignition."

2 The mixture is ignited locally, as by a spark, and a combustion wave propagates from the spark through the mixture somewhat in the manner of a light wave, that is, every point in the wave front serves as an ignition source and the propagation of the wave conforms to Huyghen's principle. This type of ignition is referred to as "ignition by a local source."

When a combustion wave has been formed in a stream of explosive gas mixture, it moves with the stream, that is, the flame blows off when the gas velocity everywhere exceeds the burning velocity, which is the velocity of the wave with respect to the unburned gas; the wave moves against the stream, that is, the flame flashes back when the burning velocity exceeds the gas velocity at any one point in the wave; and the wave remains stationary, that is, the flame is stabilized when the burning velocity equals the gas velocity at any one point and does not exceed it anywhere.

In this paper the two types of ignition and some principles of flame stabilization in a stream are discussed.

SELF-IGNITION

The self-ignition of an explosive mixture enclosed in a vessel is found to depend upon a number of variables, namely, temperature, pressure, mixture composition, and in most cases also upon gas motion and vessel factors, such as size and the condition of the inner surface of the vessel. In some cases ignition can be produced by irradiation. The phenomenon of ignition is understood to be the transition from a slow to a very rapid chemical reaction with approximately adiabatic heat evolution. Two known processes bring this about. One is self-acceleration of the reaction by the heat developed until the rate of heat liberation exceeds the rate of heat loss to the surroundings, the so-called "thermal explosion;" the other is self-acceleration caused by the formation of chemically active particles, such as atoms and free radicals, at a rate that exceeds the rate of their destruction, the so-called "branched-chain explosion."

A sharp division between these two types of explosions cannot be made, since both processes may occur simultaneously. However, a purely thermal explosion arises when the chemical reaction is of a simple order, like a unimolecular reaction for which there is evidence in the decomposition of azomethane (1)⁴ and ethyl azide (2). The quantitative treatment (3) of the temperature-time relations in the reaction vessel is based upon the equation for the rate of heat q , accumulating in the vessel, viz.

$$dq/dt = QVmAe^{-E/RT} - aK(T - T_0) \dots \dots [1]$$

where Q is the heat of reaction per mole of gas decomposed, V the volume of the reaction vessel, m the number of moles of reacting gas per unit volume, E the energy of activation, R the gas constant, a the wall area, T and T_0 the temperatures of the gas and the reaction vessel, respectively, and K and A constants; $Ae^{-E/RT} = k$, the reaction-rate coefficient. From the equation $-dm/dt = km$, and assuming k to be constant during the induction period where the temperature rise is small, $m = m_0e^{-kt}$, where m_0 is the initial molar concentration. The first term on the right side of Equation [1] is the rate of heat production, and the second term the rate of heat loss. On dividing the equation by the total heat capacity of the gas in the vessel, it becomes an equation connecting temperature with time. Its solution by numerical method leads to the result that above a certain sharply defined initial pressure a sudden temperature rise, corresponding to an explosion, occurs after an initial stage of slow temperature rise which may be regarded as the induction period. The foregoing illustrates the complexity of the quantitative treatment, even for the simplest case of a unimolecular reaction. In the

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 47-A-33.

⁴ Numbers in parentheses refer to the Bibliography at the end of the paper.

more common case of a complex reaction mechanism involving a sequence of many elementary reactions, the treatment becomes quite unmanageable. One may, nevertheless, appreciate the main features of the thermal theory by the general formulation of the explosion condition, that is, the boundary between non-explosive and explosive reaction is defined by the condition that the rates of heat liberation and dissipation are equal

$$dq/dt = QVr - aK(T - T_0) = 0 \dots \dots \dots [2]$$

r is the number of moles of gas reacting per unit time and volume, and, in most instances, is a complicated and incompletely known function of all the variables just enumerated. It is generally, however, a function of pressure in the sense that it increases with the latter, and of temperature in the form of a factor $e^{-E/RT}$. If Equation [2] has a real root in T , the system will come to equilibrium at this temperature, and the reaction will proceed normally; but if for fixed values of the other constants the pressure or T_0 is increased sufficiently, the equation will have no real root, dq/dt will be always positive, the temperature will increase, and the process of self-acceleration of the reaction, i.e., explosion, will set in. This shows that the ignition temperature T_0 is lowered by increasing pressure and also by increasing vessel diameter which increases V more than a . A plot of ignition temperature versus pressure may generally be expected to follow approximately an equation of the form $\log P = A/T_0 + B$, where A and B are constants, the latter depending, among other things, upon vessel factors. The latter equation agrees with many experimental observations, but since a similar equation holds for branched-chain explosions where the active particles are destroyed at the wall of the vessel, this general form does not serve as a criterion to distinguish between thermal and branched-chain explosions.

It has, nevertheless, been possible to draw this distinction by reviewing chemical facts and quantitative data on explosion processes. Two cases in point are the self-ignition of mixtures of hydrogen and of carbon monoxide with oxygen. Notably the former is now rather thoroughly understood. It is a branched-chain explosion with a more or less negligible contribution of the heat liberation, depending upon experimental conditions (4).

An interesting recent publication deals with the self-ignition of mixtures of gasoline-type fuels and air under conditions approximating engine knock (5). The mixtures were rapidly compressed under conditions assuring a well-defined temperature and pressure at the end of the compression, and records were obtained of the subsequent pressure rise due to heat liberation. These pressure-time records show two phases: In the first phase the pressure rises at an increasing rate for a while and then levels off. In the second phase the pressure rises from the new level at an increasing rate to explosion. The two phases respond differently to experimental variables. Tetraethyl lead is found to lengthen the second phase only. The first phase is largely independent of the fuel-air ratio over a wide range; the second phase is strongly dependent upon the fuel-air ratio. The two phases together constitute the time lag in the self-ignition of such mixtures. Their separate appearance conforms to other experience. It has long been known that mixtures of oxygen and paraffinic fuels, notably those of low octane numbers, are capable of a specific chain-branching reaction which attains a maximum rate at a fairly low temperature and, remarkably, vanishes as the temperature rises; thus the rate of fuel consumption or heat liberation initially tends toward a branched-chain explosion but the process is stopped by the temperature rise. Subsequently, a self-accelerating oxidation reaction of another type resumes the trend toward explosion, probably by a thermal mechanism. Under conditions of predominant branched-chain reaction, copious amounts of aldehydes are formed and a bluish lumines-

cence, commonly referred to as cool flame, appears in the mixture.

The initial trend toward a branched-chain explosion has been indirectly confirmed by the observation that the rate of heat liberation in the first phase is inconsistent with the thermal-explosion equation. The experimental conditions are such that the second term in Equation [1] or [2] is negligible, and it is only necessary to investigate whether the experimental values of the rate of pressure rise ($\sim dq/dt$) can be consistently represented by an equation of the form, $\text{const} \times e^{-E/RT}$, with chemically possible values of E . This was found to be not the case (5).

Engine knock is caused by self-ignition of fuel-air mixture ahead of the normal combustion wave, and the foregoing contributes considerable new information to the subject.

IGNITION BY LOCAL SOURCES

It is possible to pass small sparks through a combustible mixture without producing ignition. A comprehensive experimental and theoretical investigation of the boundary conditions which govern ignition has recently been made for electric-capacitance sparks in mixtures of methane, oxygen, and nitrogen. Because certain basic principles have developed which are applicable to other fuels as well, this work is reviewed here at some length. For further details and additional experimental data the original papers should be consulted.

In the experimental part of the investigation (6), the gases were admitted to a stainless-steel bomb at room temperature (approximately 25 C). The spark electrodes were mounted in the center of the bomb and connected to the plates of the air condenser whose capacity could be varied from 8 to 5000 μf . The experimental arrangement is shown in Fig. 1. After the

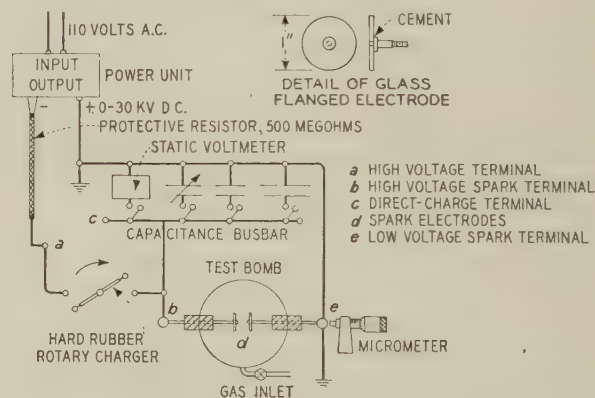


FIG. 1 SCHEME OF APPARATUS FOR DETERMINING MINIMUM ENERGIES OF IGNITING SPARKS

bomb had been charged with an explosive mixture of accurately determined composition and pressure, the condenser was slowly charged and the voltage V at which the spark occurred was observed. If no ignition occurred, the capacitance was increased and the experiment repeated until by trial and error the critical capacitance C for ignition was found. The minimum ignition energy $(1/2)CV^2$ of the circuit represents essentially the energy imparted to the gas between the electrodes in the form of heat and ionization.

It was possible to vary the breakdown voltage considerably by applying an overvoltage, either taking advantage of the breakdown time lag or using a special switch for closing the circuit. The minimum ignition energy was found to be essentially independent of overvoltage, that is, as the gap voltage was increased, the capacitance had to be correspondingly decreased.

The distance between the electrodes could be adjusted accu-

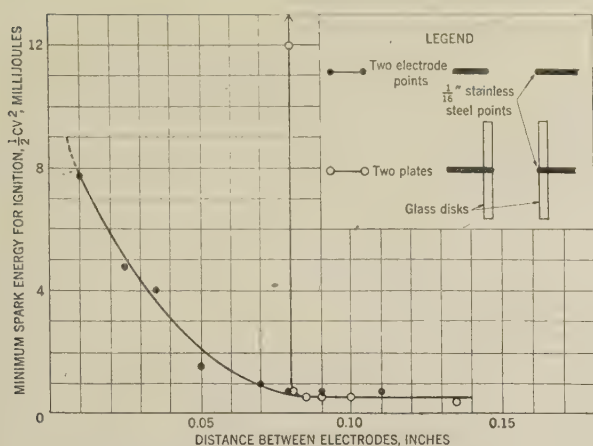


FIG. 2 MINIMUM SPARK ENERGIES FOR IGNITION FOR POINT AND PLATE ELECTRODES AS FUNCTION OF ELECTRODE DISTANCE.
(Stoichiometric mixture of natural gas and air at 1 atm pressure.)

rately by a built-in micrometer. Fig. 2 shows the effect of distance using $1/16$ -in.-diam electrodes with rounded points, either free or mounted flush in glass plates as sketched. Above a critical distance, the data practically coincide; below this distance the minimum ignition energy increases abruptly with plate electrodes and gradually with point electrodes. This distance marks the farthest penetration of the flame-quenching effect of the solid electrode material and is analogous to the critical diameter of a tube below which flame will not propagate through the tube.

As shown in Fig. 2, at distances above the quenching distance, the minimum spark energy remains remarkably constant over a considerable range. The curve turns ultimately up again; this is not shown here. The existence of a range of constant energy suggests that the spark is not incendiary along its whole length. Otherwise, as the column of gas traversed by the spark increases, the energy required to raise the gas temperature everywhere to the ignition point would also increase. Therefore it appears that most of the conversion of electrical to thermal energy occurs at some point along the spark path. This accords well with the known charge distribution during discharges in gases. The largest decrease of potential along the spark path occurs near the cathode due to the accumulation of positive ions. The high concentration of positive ions and fast electrons at the end of this zone of steep potential drop must result in a dissipation of energy much larger than anywhere else along the path of the spark. Thus a virtual point source of ignition is formed independent of the length of the spark.

One may further visualize that as the electrode distance is decreased, the ignition source approaches the electrodes to within a critical distance and the flame is quenched—abruptly in the case of plate electrodes and gradually with point electrodes. If the source is always closer to the cathode than to the anode, then, in the case of a point electrode facing a plate electrode, the quenching effect should be larger when the plate is the cathode than when it is the anode. This has been confirmed by experiments in which one of the electrodes was a small sphere and the other a plate.

In the constant range of the electrode-distance curve the minimum spark energy is dependent only upon the variables of the explosive gas mixture. The minimum spark energy and the quenching distance increase with decreasing pressure and pass through a minimum when the methane percentage is increased from the lower to the upper limit of inflammability. Typical curves are shown in Fig. 3. Families of curves of minimum ignition energy versus methane percentage at constant ratios of

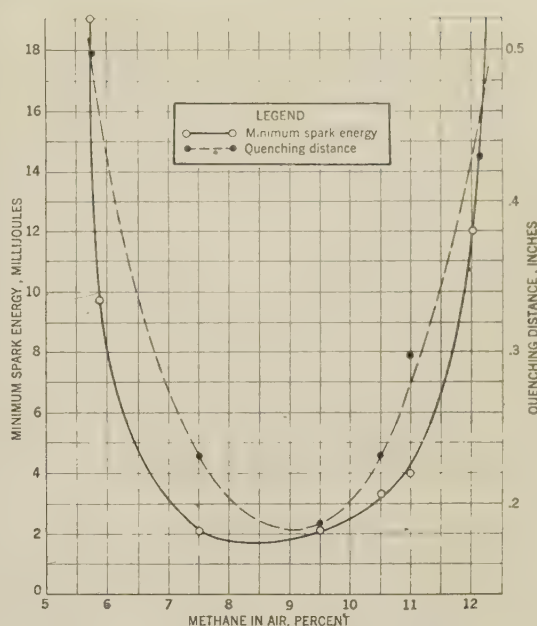


FIG. 3 MINIMUM SPARK ENERGIES AND QUENCHING DISTANCES OF MIXTURES OF METHANE AND AIR AT $1/3$ ATM PRESSURE
(Point electrodes embedded in glass plates.)

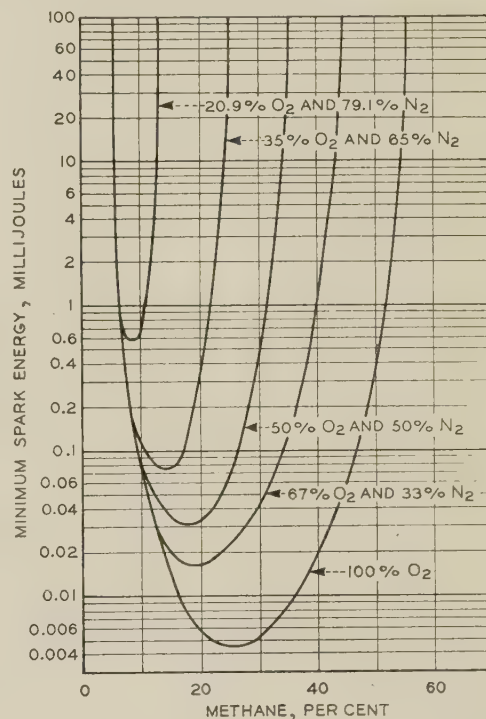


FIG. 4 MINIMUM SPARK-IGNITION ENERGIES OF MIXTURES OF METHANE, OXYGEN, AND NITROGEN AT 1 ATM PRESSURE
(Curves correspond to constant oxygen:nitrogen ratio. Point electrodes embedded in glass plates.)

nitrogen to oxygen have been determined for various pressures. Fig. 4 shows the curves for 1 atm pressure. They illustrate the profound effect of inert-gas dilution. Comparing, for example, measurements for air and pure oxygen near stoichiometric composition, the minimum spark energy of the latter mixture is found to be about one hundredth of the former.

The theoretical part of the investigation (7) is based upon considerations of the distribution of temperature and energy in a

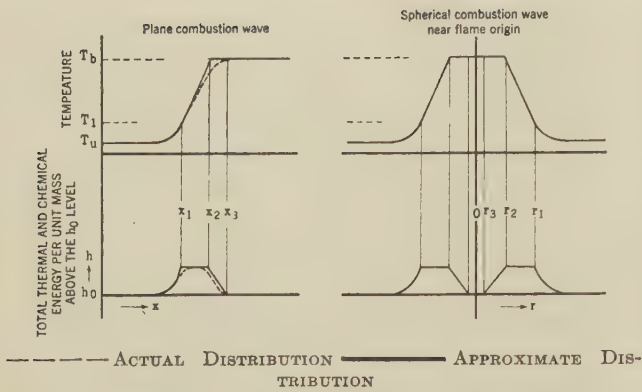


FIG. 5 DISTRIBUTION OF TEMPERATURE AND ENERGY PER UNIT MASS IN A COMBUSTION WAVE

(h_0 = sum of thermal and chemical energy per unit mass before and after combustion.)

combustion wave, as illustrated in Fig. 5. For the case of a plane wave the origin of the x -co-ordinate is placed at a constant distance from the wave, and, correspondingly, the gas flows from left to right at the velocity of flame propagation. At any point x , the sum of the rates of heat gain or loss in a unit volume of a layer dx due to thermal conduction, mass flow, and chemical reaction is zero. This is expressed by the following equation

$$\mu d^2T/dx^2 - \rho S c_p dT/dx + q(x) = 0 \dots \dots \dots [3]$$

where

- T = temperature
- μ = heat-conductivity coefficient
- ρ = density
- S = velocity of gas flow
- c_p = specific heat
- $q(x)$ = rate of heat release by chemical reaction per unit volume

The temperature rises from T_u of the "unburned" gas to T_b of the "burned" gas. The chemical term $q(x)$ is initially very small; hence according to Equation [3], $d^2T/dx^2 \sim dT/dx$, which means that the temperature gradient and also the temperature increase exponentially with x , until at some temperature T_1 the chemical term becomes significantly large; d^2T/dx^2 now decreases and finally becomes negative, corresponding to the dotted temperature curve in Fig. 5. Accordingly, as shown in the lower part of the figure, a mass element entering the combustion wave first acquires some excess energy by heat conduction from preceding elements and then returns this "borrowed" energy to succeeding elements; so that, as demanded by thermodynamics, its thermal energy after passage through the combustion wave is equal to the sum of its thermal and chemical energy before it entered the combustion wave. In the aggregate, the mass elements inside a plane wave segment of unit area carry an excess energy

$$\int_0^{+\infty} \rho(h - h_0) dx$$

where h represents the sum of thermal and chemical energy per unit mass and the subscript zero refers to the state before and after passage through the combustion zone.

The initial small flame which is formed around a spark derives its excess energy from the spark itself. It is the function of the latter to initiate the reaction by producing a high local concentration of heat and chain carriers, and to furnish at least as much energy as is necessary to satisfy the excess-energy requirement of the smallest flame sphere that is capable of self-propagation. Such a flame may be visualized as a burning sphere of the smallest

volume consistent with the condition that the rate of heat production should equal the rate of heat conduction to the surrounding unburned gas. As the flame grows, the additional energy required by the growing flame surface is taken from the burned gas, that is, the combustion temperature is slightly lower than the temperature T_b of the plane wave. However, the spark must furnish at least the energy corresponding to the integral

$$\int_0^{\infty} 4\pi r^2 \rho(h - h_0) dr$$

of the smallest self-propagating flame, r being the distance from the center of the flame.

The calculation of this energy from other data, notably data on the burning velocity and the diameter of the smallest flame, is made possible by substituting for the actual temperature gradient the simple approximate gradient illustrated in Fig. 5. The term $q(x)$, Equation [3], is taken as zero up to the temperature T_1 , and thus the gradient initially rises exponentially. Between T_1 and T_b , $q(x)$ is taken as constant and d^2T/dx^2 as zero, and the gradient thus remains constant. A mass element within this part of the gradient neither loses nor gains h = thermal + chemical energy. However, heat is continually being furnished to the fresh gas at x_1 ; this heat is being produced at the constant temperature level T_b from residual chemical energy between x_2 and x_3 , and transported down the gradient by a fictitious mechanism. Although the resulting temperature and energy distributions are not physically possible, they closely circumscribe the actual distributions. They can be calculated numerically from data on the width of the reaction zone $x_3 - x_1$, the burning velocity S_u , the temperatures T_u and T_b , and the heat conductivity, density, and specific heat of the unburned gas.

For the present problem of calculating the minimum ignition energy, the heat-balance equation for a spherical combustion wave must be developed, using the foregoing approximation. The development of the equations has been given elsewhere (7). The solution of the differential equations yields the temperature distribution in the exponential part of the gradient at radii greater than r_1 . This makes it possible to solve the equation for the excess energy in this part of the gradient.

$$\left| H \right|_{r_1}^{\infty} = \int_{r_1}^{\infty} 4\pi r^2 \rho(h - h_0) dr = \int_{r_1}^{\infty} 4\pi r^2 \rho c_p (T - T_u) dr \dots [4]$$

The energy between r_2 and r_1 is given by

$$\left| H \right|_{r_2}^{r_1} = \int_{r_2}^{r_1} 4\pi r^2 \rho c_p (T_1 - T_u) dr \dots \dots \dots [5]$$

It can readily be shown that the value of $\left| H \right|_{r_2}^{r_1}$ is less than one third that of $\left| H \right|_{r_1}^{\infty}$. The excess-energy term $\left| H \right|_0^{r_2}$ is considerably less than $\left| H \right|_{r_1}^{r_2}$ and therefore can be neglected.

The gradient $(T_b - T_1)/(r_1 - r_2)$ is stable and thus the flame can propagate when the rate of heat production in the volume $(4/3)\pi r_2^3$ equals the heat flow across the area $4\pi r_1^2$. From this condition a relation between r_1 and r_2 is derived and the gradient between r_1 and r_2 can be calculated. This furnishes the necessary boundary condition for the integration of Equation [4]. It now becomes possible to write the equation of the ignition energy H , which is found to be

$$H = 4\pi r_1^2 \frac{\mu_u}{S_u} \sqrt{T_u/T_u} (T_b - T_u) \left(\frac{a_{\infty}}{a} \right)^2 \frac{1 + 1.3 ar_1}{1 + a_{\infty} r_1 (1 - \sqrt[3]{3/ar_1})} \dots [6]$$

where S_u is the burning velocity determined for a plane com-

bustion wave and μ_u the coefficient of heat conductivity of the unburned gas. T_a is the average temperature between T_u and T_1 and is found from the condition that the energy

$$\left| H \right|_{r_1}^{r_2} = \left| H \right|_{\infty}^{r_2}$$

The function a_{∞} is given by

$$a_{\infty} = (c_p(u)\rho_u S_u/\mu_u) \sqrt{T_u/T_a} \dots \dots \dots [7]$$

The relation between a_{∞} and a is

$$1/a_{\infty} r_1 = 1 - ar_1 e^{ar_1} \int_{r_1}^{\infty} \frac{e^{-ar}}{r} dr \dots \dots \dots [8]$$

Burning velocities of mixtures of methane and oxygen and nitrogen have been determined by the Bunsen-burner method (8). Values for c_p and μ and ρ are available in handbooks. Values of the flame temperature T_b are calculated from thermodynamic data.

The remaining unknown quantity in Equation [6] is r_1 . The diameter $2r_1$ evidently must be smaller than the quenching distance d between plane parallel electrodes mentioned in the experimental part of this paper. This follows immediately from inspection of the temperature distribution in a spherical combustion wave, Fig. 5. Since, for minimum ignition conditions, none of the excess energy may be removed, it is obvious that the cooling surfaces must be well outside the exponential part of the temperature gradient. It is difficult to state how great this distance should be. It is not sufficient to note where the temperature has dropped to nearly T_u because considerations of the quenching mechanism suggest that a plate distance which may be sufficient for the initial development of the flame may result in quenching at a later stage of flame development. A reasonable estimate of the ratio $d/2r_1$ is ~ 2 .

Equation [6] has been used to calculate values of $2r_1$ from the measured minimum ignition energies, and these values are compared with measured values of d . Results are given in Table 1. The examples chosen include a wide range of mixtures of methane, oxygen, and nitrogen. A reasonable correspondence between d and $2r_1$ is noted.

Equation [6] is of such a form that H , as a function of the radius r_1 , passes through a minimum. This is a consequence of the approximating assumptions embodied in the treatment and cannot be considered physically significant. In some cases this calculated minimum value is larger than the experimental value of H , and it becomes impossible to satisfy the experimental value by any choice of r_1 . It then appears proper to solve the equation for the minimum value, as this is the closest approach to the experimental value of H . It is noted in Table 1 that this was done in three cases corresponding to low methane percentages. The discrepancies are not serious.

The treatment thus establishes a concept of the minimum

ignition energy of sparks which links this quantity with the variables of flame propagation, in agreement with experimental data. However, it may be noted that in these experiments the spark energy is the sole minimum ignition requirement only because (a) the high concentration of energy in the sparks insures rapid chemical reaction, and (b) the time of $\sim 10^{-7}$ sec, during which the energy of a capacitance spark is released, is small compared with the time required for a substantial growth of the initial flame. For example, in the case of the fast-burning mixture of 25 per cent methane and 75 per cent oxygen, a 50 per cent increase of the flame radius r_1 requires $\sim 10^{-5}$ sec, according to the values of S_u and r_1 in Table 1. Future experiments with greatly increased duration of the electric discharge should demonstrate a dependence of the minimum ignition energy upon the discharge time.

Local ignition by hot wires and heated surfaces, in general, represents a very different picture. Experimentally, it is usually possible to determine a minimum ignition temperature. Its value depends upon many variables and is often much above the temperature range in which self-ignition in heated vessels occurs. For example, mixtures of methane and air, which ignite spontaneously in vessels below 750 C, require local ignition temperatures generally above 1000 C and up to 1500 C, depending upon many factors, such as the nature of the heated surface and its vertical or horizontal position (9).

FLAME STABILIZATION (10)

Consider an explosive gas mixture in a tube whose diameter is sufficiently large to permit the propagation of a combustion wave. The wave cannot approach the wall of the tube closer than the quenching distance. This distance is a function of the composition of the mixture, the temperature, and the pressure. It is somewhat smaller than one half the quenching distance between plane parallel plates. The burning velocity is zero within the quenching distance and at larger distances increases to its normal value. This is shown schematically by the heavy curve in Fig. 6. The other lines in the figure are curves of gas velocity for three different gas flows. The gas velocity is zero at the wall and rises toward a maximum in the axis of the stream. If the flow corresponds to curve 1, the gas velocity falls partly below the burning velocity, and the flame flashes back through the tube. If the flow corresponds to curve 2, the combustion wave remains stationary in an unstable equilibrium position. If the flow corresponds to curve 3, the gas velocity is everywhere larger than the burning velocity, and the flame is blown out of the tube.

In the latter case, as the combustion wave propagates in the free stream beyond the rim of the tube, the quenching effect of the tube rim gradually vanishes, and, consequently, the curve of the burning velocity shifts closer toward the stream boundary. This is illustrated in Fig. 7. Three burning-velocity curves are shown corresponding to three heights, A, B, and C above the rim. At height A, close to the rim, the burning-velocity curve

TABLE 1. CALCULATION OF FLAME DIAMETERS $2r_1$, TEMPERATURES T_b , T_1 , AND T_a , AND GRADIENTS $(T_b - T_1)/(r_1 - r_2)$ FOR SMALLEST IGNITING FLAMES. COMPARISON OF FLAME DIAMETERS WITH FLAME-QUENCHING DISTANCES d BETWEEN PARALLEL PLATES; $T_u = 300$ DEG K

Mixture composition, — per cent							Minimum ignition energy H , cal. $\times 10^6$		$2r_1$, cm	d , cm	$d/2r_1$	$(T_b - T_1)/$ $(r_1 - r_2)$, deg C/ cm $\times 10^6$
CH ₄	O ₂	N ₂	S_u , cm/sec	Deg K			Experi- mental	Calcu- lated ^a				
				T_b	T_1	T_a						
10	90	...	80	2200	1330	587	19	36	0.038	0.072	1.9	3.5
15	85	...	175	2650	1570	658	3.1	3.9	0.017	0.050	2.9	9.5
25	75	...	304	3000	822	515	1.4	...	0.016	0.028	1.8	8.0
40	60	...	305	3000	605	435	5.0	...	0.032	0.051	1.6	3.5
50	50	...	122	2650	446	379	110	...	0.106	0.165	1.6	0.8
52.5	47.5	...	87	2500	388	343	430	...	0.192	0.38	2.0	0.38
10	18.8	71.2	36	2200	1115	548	170	382	0.090	0.28	3.1	1.3
16.3	29.3	54.4	110	2620	1100	567	20	...	0.035	0.079	2.3	3.8
21.5	39.75	39.75	170	2810	791	501	9	...	0.031	0.053	1.7	3.9
26.4	49.1	24.5	240	2940	592	429	5.3	...	0.030	0.045	1.5	3.6

^a Values obtained by minimizing Equation [6].

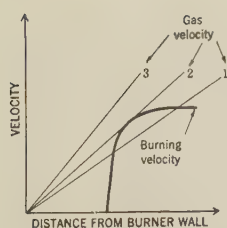


FIG. 6 BURNING VELOCITY AND GAS VELOCITY INSIDE BURNER TUBE

(Curve 1, flash back; curve 2, flash-back limit; curve 3, no flash back.)

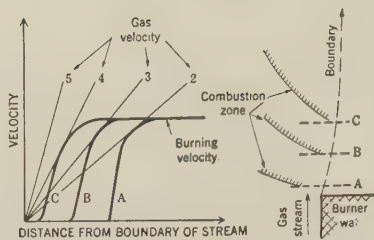


FIG. 7 BURNING VELOCITY AND GAS VELOCITY ABOVE TUBE RIM

(Curves A, B, C; burning velocity at heights A, B, C. Curve 2, flash-back limit; curve 3, stable flame; curve 4, blowoff limit; curve 5, blowoff.)

is about the same as in the diagram, Fig. 6. At height B, it has shifted toward the boundary. The shift toward the boundary continues up to height C. Here the quenching effect of the tube rim is very small, but the curve drops to zero at the stream boundary because by interdiffusion with air and transfer of momentum an outermost layer of nonexplosive gas is formed. At heights exceeding C the nonexplosive boundary layer broadens, and correspondingly the burning-velocity curve recedes from the boundary; hence if the gas velocity is large as shown in curve 5, it exceeds the burning velocity everywhere and the flame blows off the tube. At any flow between the limiting curves 2 and 4, the flame settles down to such a height above the rim that gas-velocity curve and burning-velocity curve meet each other tangentially. For example, let us suppose that the gas stream is adjusted to correspond to the gas-velocity curve 3. If the combustion wave drops below the height B, the burning-velocity curve shifts to the right, the gas velocity is larger everywhere, and the combustion wave lifts up again toward the height B. If it should exceed this height, the burning-velocity curve shifts to the left, the gas velocity falls below the burning velocity at some distance from the boundary, and the combustion wave is driven back to its equilibrium position at height B.

The flame thus remains stable between a critical lower and upper gradient of the gas velocity at the stream boundary, represented by the slopes of curves 2 and 4, and corresponding to

flash back and blowoff, respectively. For a flame burning in air, the blowoff gradient increases sharply when the mixture is enriched with fuel gas, because in this case the interdiffusing air increases the burning velocity at the boundary. Therefore rich flames are much more stable. However, if the surrounding atmosphere does not consist of air, but of some inert gas, such flames blow off readily.

If the velocity distribution in the gas stream is known, the critical boundary-velocity gradients can be calculated from data on the critical flows for flash back and blowoff. For laminar flow in a cylindrical tube of radius R , the velocity U at the distance r from the axis is given by

$$U = -(a/4\eta)(R^2 - r^2) \dots \dots \dots [9]$$

where η is the viscosity and a the hydrodynamic pressure gradient along the tube. The flow V is

$$V = \int_0^R 2\pi U r dr = - (a/4\eta)(\pi/2)R^4 \dots \dots \dots [10]$$

From Equations [9] and [10] the boundary velocity gradient g is found to be

$$g = \lim_{r \rightarrow R} (-dU/dr) = 4V/\pi R^3 \dots \dots \dots [11]$$

Fig. 8 shows critical flows for flash back of natural gas-air flames in cylindrical tubes of various diameters. Fig. 9 shows the same data plotted against g instead of V . In this plot the curves coincide substantially, as expected. Exceptions are noted whenever the tube diameters approach the limiting diameters for flame propagation. This is to be expected, because the quenching distances are no longer small compared to the tube radii and therefore the gas velocity does not increase linearly over the quenching distance, that is, the concept of a critical boundary-velocity gradient is no longer applicable. Another exception is the notable protrusion of the curve for the largest tube (1.550 cm diam). This is caused by the asymmetric distribution of the gas velocity induced by the thrust of the flame gas, as illustrated by an example shown in Fig. 10. Here the flow exceeds the critical flow for flash back but the combustion wave has partly entered the tube. The thermally expanding

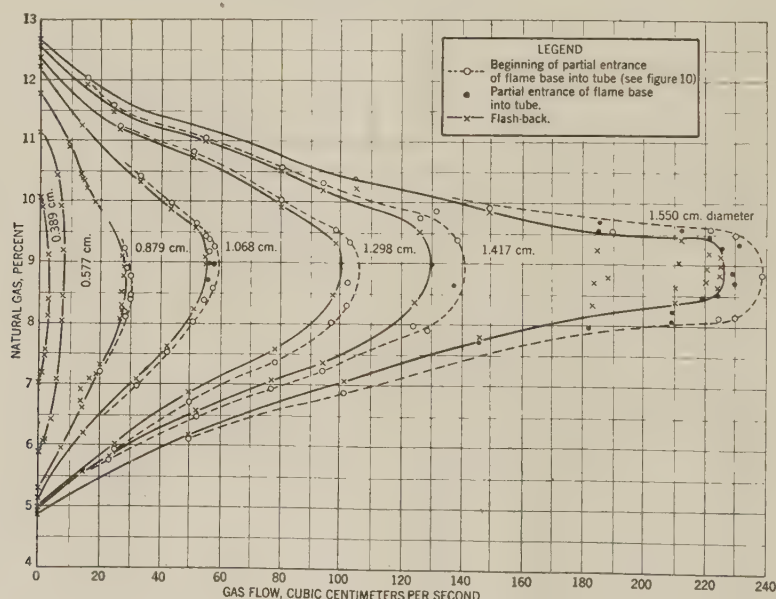


FIG. 8 CRITICAL FLOWS FOR FLASH BACK OF NATURAL GAS-AIR FLAMES IN CYLINDRICAL TUBES OF DIFFERENT DIAMETERS AT ROOM TEMPERATURE

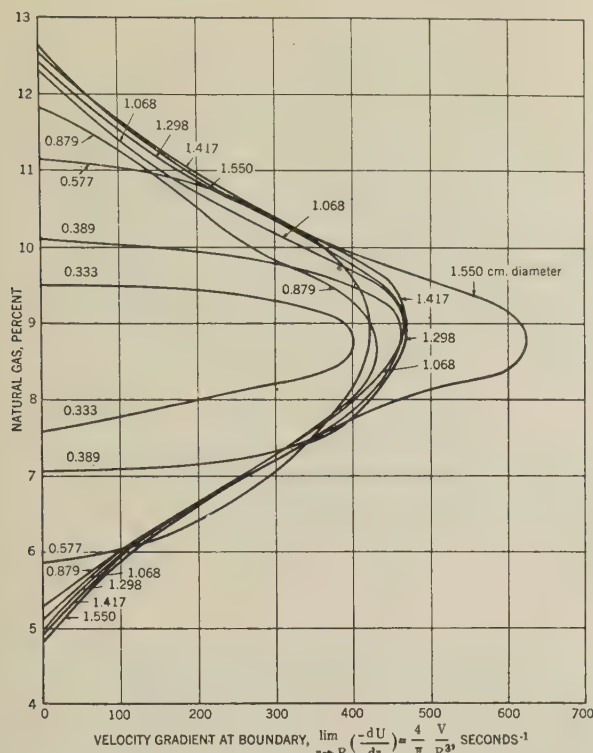


FIG. 9 CRITICAL VELOCITY GRADIENTS AT BOUNDARY OF GAS STREAM FOR FLASH BACK OF NATURAL GAS-AIR FLAMES IN CYLINDRICAL TUBES OF DIFFERENT DIAMETERS AT ROOM TEMPERATURE

flame gas exerts a thrust normal to the tube axis which causes a deflection of the stream of unburned gas. The resulting change of velocity distribution causes a decrease of the boundary-velocity gradient, below the critical flash-back value, in the region where the flame penetrates into the tube; this permits the combustion wave to propagate against the stream. As the wave travels downward, the resistance of the flow of unburned gas to the deflecting thrust increases, due to the increasing confinement by the tube wall, and a position of equilibrium is reached in which the boundary velocity gradient at the deepest point of penetration has become equal to the critical gradient for flash back, but exceeds the critical value at lower points in the burner tube. The tendency toward tilted flames of this type increases with increasing tube diameter, and the flame is not always arrested in an equilibrium position but flashes back with an irregular wave front, indicating induced turbulence in the stream. The flash-back region may thus be increased beyond the normal boundary, as exemplified in Fig. 9.

Observations of blowoff in the laminar-flow range have shown no appreciable effect of the tube diameter on the critical boundary velocity gradient, and the stability range of laminar flames of natural gas-air mixtures burning in air is essentially circumscribed by the curves in Fig. 11. If the flames are surrounded by some other gas instead of air, the flash-back curve remains unchanged, but the blowoff curve is shifted considerably. Oxygen on the outside greatly enlarges the stable-flame region, and inert gases, notably helium, greatly reduce the region.

The general considerations just cited are evidently applicable as well to an inner stream boundary, as, for example, the surface of a wire mounted axially in a cylindrical tube. Fig. 12 shows a flame stabilized by such a wire while blowing off at the outer stream boundary. For laminar flow, the velocity gradient at the wire surface can be calculated from the equation of flow through an annular channel (11). Observations of blowoff from wires

are summarized in Fig. 13. The critical velocity gradients are constant over a considerable range of wire diameters. For very fine wires they become smaller, and for very large wires they become larger, as may be expected from additional considerations of the modification of the gas flow above the wire.

The effect of dilution or enrichment by the surrounding atmosphere does not enter into the blowoff from an inner stream boundary. Hence for lean mixtures, the "inner" blowoff gradi-

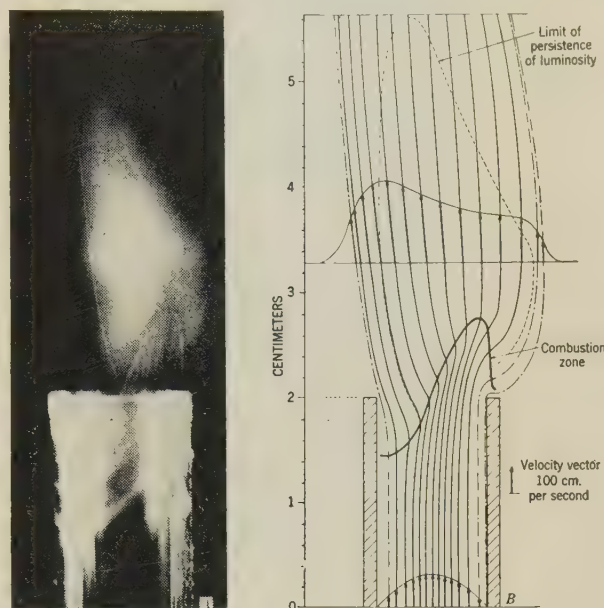


FIG. 10 FLAME OF NATURAL GAS AND AIR, PARTIALLY DRAWN INTO BURNER TUBE AT GAS FLOW JUST EXCEEDING CRITICAL FLOW FOR FLASH BACK

(Mixture composition, 8.1 per cent natural gas in air; gas flow, 52 cu cm per sec; inside diameter of tube, 1.068 cm. A, Particle tracks in vertical center plane. B, Diagram of flow pattern and velocity distribution. Solidly drawn flow lines are so located that mass flow between any two adjoining lines is the same.)

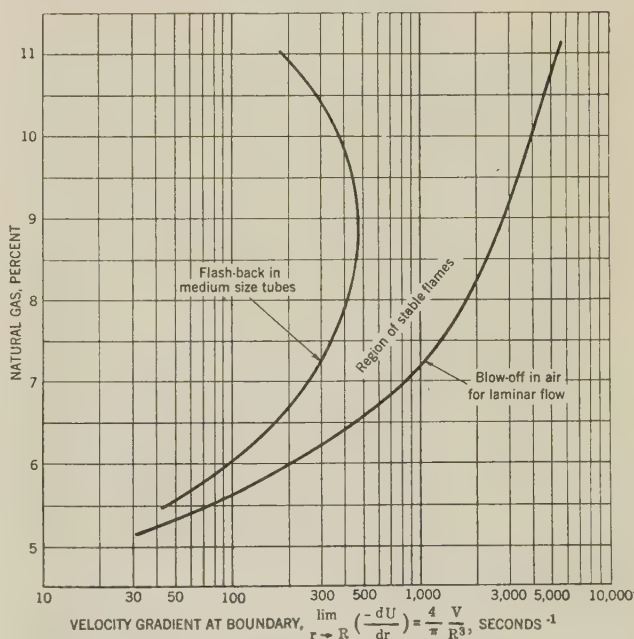


FIG. 11 REGION OF STABLE NATURAL GAS-AIR FLAMES FOR LAMINAR STREAMS IN CYLINDRICAL BURNER TUBES

ents are larger than the "outer" ones, while for rich mixtures the reverse is true.

The applicability of the concept of critical boundary velocity



FIG. 12 FLAME ANCHORED IN CENTER OF STREAM OF COMBUSTIBLE GAS BY MEANS OF AXIALLY MOUNTED WIRE, FORMING INVERTED CONE

(Interrupted lines are tracks of fine magnesium-oxide particles, made visible by stroboscopic illumination to show direction and velocity of streamlines.)

gradients to the turbulent-flow range has been recently demonstrated on flames of propane and air (12).

BIBLIOGRAPHY

- 1 "The Explosion of Azomethane," by A. O. Allen and O. K. Rice, *Journal of the American Chemical Society*, vol. 57, 1935, pp. 310-317.
- 2 "The Explosion of Ethyl-Azide," by H. C. Campbell and O. K. Rice, *Journal of the American Chemical Society*, vol. 57, 1935, pp. 1044-1050.
- 3 "The Induction Period in Gaseous Thermal Explosions," by O. K. Rice, A. O. Allen, and H. C. Campbell, *Journal of the American Chemical Society*, vol. 57, 1935, pp. 2212-2222.
- 4 "Mechanism of the Thermal Reaction Between Hydrogen and Oxygen," by G. von Elbe and B. Lewis, *Journal of Chemical Physics*, vol. 10, 1942, pp. 366-393.
- 5 "Investigations on Knocking Combustion," by H. Roeger; "The Self-Ignition of Mixtures of Hydrocarbon and Air Subjected to Very Sudden Adiabatic Compression," by W. Jost (German). U. S. Government Technical Oil Mission (TOM) microfilm reel 242. Order by reel number from Photoduplication Service, Library of Congress, Washington 25, D. C. W. Jost's paper is also printed as Field Information Agency, Technical (US) (FIAT) Report No. 873.
- 6 "Ignition of Explosive Gas Mixtures by Electric Sparks, I. Minimum Ignition Energies and Quenching Distances of Mixtures of Methane, Oxygen, and Inert Gases," by M. V. Blanc, P. G. Guest, G. von Elbe, and B. Lewis, *Journal of Chemical Physics*, vol. 15, 1947, pp. 798-802.
- 7 "Ignition of Explosive Gas Mixtures by Electric Sparks, II. Theory of the Propagation of Flame From an Instantaneous Point Source of Ignition," by B. Lewis and G. von Elbe, *Journal of Chemical Physics*, vol. 15, 1947, pp. 803-808.
- 8 "Der Zündvorgang in Gasmischungen," G. Jahn, Oldenbourg Berlin, Germany, 1934.

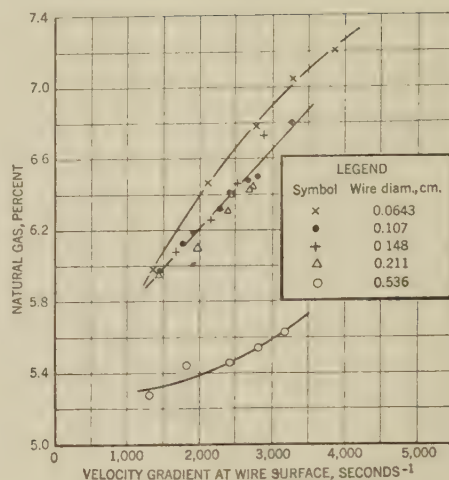


FIG. 13 CRITICAL VELOCITY GRADIENTS AT SURFACES OF WIRES FOR BLOWOFF OF INVERTED NATURAL GAS-AIR FLAME CONES FROM ENDS OF WIRES MOUNTED IN AXIS OF A CYLINDRICAL TUBE OF 1.417 CM DIAM

9 "Ignition of Natural Gas-Air Mixtures by Heated Surfaces," by P. G. Guest, Bureau of Mines Tech. Paper 475, 1930.

10 "Stability and Structure of Burner Flames," by B. Lewis and G. von Elbe, *Journal of Chemical Physics*, vol. 11, 1943, pp. 75-97.

"Further Studies of the Structure and Stability of Burner Flames," by G. von Elbe and M. Mentser, *Journal of Chemical Physics*, vol. 13, 1945, pp. 89-100.

11 "Hydrodynamics," by Horace Lamb, Cambridge University Press, London, England, fifth edition, 1924, p. 555.

12 "Experiments on Stability of Bunsen Burner Flames for Turbulent Flow," by L. M. Bollinger and David T. Williams, National Advisory Committee for Aeronautics, Technical Note 1234, June, 1947.

Discussion

J. P. LONGWELL.⁵ The work reported on ignition sources gives quantitative information on the energy required for ignition that is of considerable value in understanding this process. The progress made in describing mathematically the stability of simple systems such as the spark-ignition apparatus and the Bunsen burner by setting up idealized systems has given a better understanding of combustion apparatus. It is to be hoped that it will soon be possible to describe more rigorously these relatively simple systems and to apply similar methods of analysis to the more complex systems encountered in practice. The thickness of the flame front used in the analysis of the minimum energy required for spark ignition is an important part of this analysis and is of considerable interest in itself. It would be appreciated if the authors would give a reference to the source of the information on flame thickness.

It is hoped that studies such as those made by the authors can be used to develop tests which will rate fuels used in jet-propulsion equipment according to their ease of ignition and combustion, so that the relative performances of fuels in various types of combustion equipment can be predicted. It would be desirable that this test measure some basic quantity which depends only upon the composition, pressure, and temperature of the mixture and is independent of the particular test equipment. Measurements of such quantities as burning velocity or minimum spark-ignition energy could possibly serve this purpose. Examination of the data indicates that a large increase in flame velocity is reflected in a decrease in minimum spark-ignition energy and mini-

⁵ Process Division, Esso Laboratories, Linden, N. J.

mum plate spacing. While this correlation was rough it indicated, as did the analysis, that these three quantities are not independent, and probably do not indicate entirely different characteristics for purposes of rating fuels. It is hoped that data on the hydrocarbons used in jet-propulsion practice will be obtained so comparisons can be made with the performance of these fuels in more complex combustion equipment.

A related study is being carried out by the Standard Oil Development Company partially supported by the Office of Naval Research. In this work various-size capacitors, charged to 600 volts, are discharged through a spark gap of 2.8 mm in mixtures of hydrocarbon, nitrogen, and oxygen. The present phase of this work is concerned with ignition at very low pressures, and Table 2 of this discussion shows the minimum pressure at which ignition could be obtained for various capacitances.

TABLE 2 CAPACITANCE REQUIRED FOR IGNITION OF HYDROCARBON AIR MIXTURES AT LOW PRESSURES

Capacitance, mfd	Minimum pressure at which ignition can be obtained, mm Hg		
	Iso-octane	N-Butane	Butadiene
4	44	46	26
8	37	34	26
18	31	28	21
48	31	28	21

These mixtures were all slightly richer than stoichiometric, and it is indicated that considerably larger capacitors are needed at these low pressures than were used in the Bureau of Mines work at higher pressures. It is also noted that increasing the capacitance beyond 18 mfd did not allow ignition at lower pressures, so that this pressure might be taken as the minimum pressure at which ignition can be obtained with the particular hydrocarbon-air mixture. The effect of addition of oxygen in making ignition easier is demonstrated by the fact that, in these tests, ignition was possible at less than 3 mm pressure with mixtures of butane and oxygen. Studies at such low pressures are made difficult by the fact that the spark discharge becomes quite diffuse.

G. C. WILLIAMS.⁶ The present paper exhibits the carefully performed experiments and thorough attempt at elucidation of experiment in terms of theoretical analysis characteristic of the authors' writings in this field. Because of the summarizing nature of the present paper, several questions which appear to be rather incompletely answered in this presentation may, in effect, be answered by reference to more detailed supporting articles in another journal (authors' references 6 and 7).

In the description of the function of the excess-energy concept for the spreading combustion wave, it is clearly stated that a number of assumptions are made to arrive at a manageable mathematical relation between minimum ignition energy H and minimum flame radius r_1 . No mention, however, is made of assumptions regarding the importance of heat flow by the mechanism of radiation. Allowance could be made for this mode of heat transfer by proper choice of a fictitious thermal conductivity μ for the gas. However, this would entail different fictitious values of μ for every flame temperature and surroundings temperature. In addition, it can be demonstrated mathematically that if radiation from the flame front is sufficiently important, the gases at some point within the flame front can be at a higher temperature than that corresponding to the adiabatic equilibrium flame temperature, and the temperature and enthalpy curves in Fig. 5 of the paper would have to be modified to allow for heat reception by nonluminous burned gases. The question can be raised as to appropriateness of the use of burning velocity S_u for a plane combustion wave in a relation for flame propagation in systems having

such small radii of curvature. It appears probable that a lower value would obtain for propagation from a "point" than for planar propagation. Whether this difference would be of no greater magnitude than that introduced by the approximations acknowledged made may be worth investigating.

Presumably the data given in Table 1 of the paper are for experiments made at 1 atm. A statement of the test pressure or of values of q for each of the tests would be enlightening. Apparently both minimum ignition energy and quenching distance increase with decrease in pressure. Comparison of Figs. 3 and 4 of the paper indicates that for a given gas composition the proportionality of minimum energy with pressure is approximately inverse. It would be of interest to know the effects of mixture pressure and temperature on both the variables H and r_1 .

The statement regarding the effect of spark-discharge time on minimum ignition energy brings to mind another rather practical question: Are there any data on the effect of mixture velocity flowing past a spark gap on the minimum energy or quenching distance? It would appear offhand that the former, if sufficiently rapid in discharge, might be relatively unaffected unless turbulence scales of the order of, or smaller than, $2r_1$ were involved; whereas the latter, if measured by the spark gap, might be considerably reduced by increased velocity.

The statement that blow-off gradients for "inner" boundaries are less than those for "outer" boundaries in the fuel-rich region does not appear to be borne out by any experimental data in the papers quoted. In any event, application of the concept of critical inner-boundary velocity gradients should certainly be made carefully to systems where heat transfer from the flame to the boundary and thence to the fresh gas can result in considerable preheating of the gas mixture before arrival at the flame front. In such cases the thermally conducting axial length of the flame anchor may be quite as important as the developed boundary layer in stabilization of inverted flames.

AUTHORS' CLOSURE

We have estimated the temperature changes in the combustion wave due to radiation, using values of emissivity and absorptivity of water vapor and carbon dioxide given by Hottel and Mangelsdorf,⁷ also Hottel and Smith,⁸ and find that they cannot exceed a few hundredths of a degree. This estimate was arrived at by considering the energy absorbed by a layer of pure water vapor (whose absorptivity coefficient is considerably larger than that of combustion gases) of the thickness of the combustion wave, during the time of passage through the wave. The layer is assumed to be exposed to the radiation of a black body at the flame temperature.

The burning velocity in a plane combustion wave is proportional to the function a_∞ , defined by Equation [7] of the paper. The burning velocity for any radius of curvature is smaller by a factor a/a_∞ . The value of a is found from Equation [8].

The data given in Table 1 are for burning-velocity determinations at atmospheric pressure; by an oversight mention of this was omitted. Bunsen-burner measurements of burning velocity have thus far been carried out only at atmospheric pressure.

The minimum ignition energy and quenching distance increase with decreasing pressure. Details of these results are given in the paper referred to in reference (6).

There are no data on the effect of gas velocity on the minimum

⁶ Department of Chemical Engineering, Massachusetts Institute of Technology, Cambridge, Mass.

⁷ "Heat Transmission by Radiation From Non-Luminous Gases. II—Experimental Study of Carbon Dioxide and Water Vapor," by H. C. Hottel and H. G. Mangelsdorf, Trans. AICHE, vol. 31, 1934-1935, pp. 517-549.

⁸ "Radiation From Nonluminous Flames," by H. C. Hottel and V. C. Smith, Trans. ASME, vol. 57, 1935, pp. 463-470.

ignition energy and quenching distance. As Professor Williams points out, one should expect for sufficiently rapid discharge no effect on the minimum ignition energy unless turbulence scales of the order of, or smaller than, $2r_1$ were involved. One may visualize, in principle, that with sufficiently high gas velocity the flame will clear the quenching surfaces, assuming the latter to consist of small plates parallel to the flow.

It is true that no data are given in the paper to substantiate the statement that blowoff gradients for inner boundaries are less

than those for outer boundaries in the fuel-rich region. However, as the limit of inflammability is reached, burning velocities become very small, hence the critical velocity gradient for inner boundary blowoff must vanish, while the gradient for outer boundary blowoff remains large owing to diffusion of atmospheric oxygen into the outer boundary. This can be illustrated simply by decreasing gradually the air supply to an ordinary Méker burner. As the mixture becomes rich the small inner flame cones blow off, giving way to one large cone anchored to the rim.

Gas-Turbine Plant Combustion-Chamber Efficiency

By A. L. LONDON,¹ STANFORD UNIVERSITY, CALIF.

Concepts of gas-turbine plant combustion-chamber efficiency are considered. Three distinct efficiency terms are required, one applying to incomplete combustion and setting losses, one to the flow friction and auxiliary-drive energy requirements, and the third to the influence of these losses on over-all plant thermal efficiency and output. These efficiencies are defined and details of application indicated. The proposed concepts are recommended for consideration as standards since they are thermodynamically meaningful, unambiguous, and relatively easy to apply both in evaluation of test performance and in cycle analysis.

NOMENCLATURE

- c_p, c_v = gas unit heat capacities at constant pressure and volume, respectively, Btu/(lb deg F)
 E = combustion-chamber auxiliary-drive energy requirement, Btu/lb of turbine flow
 f, f_i = fuel requirement of the actual and ideal combustion chambers, respectively, lb/lb of air supply
 h = unit thermal enthalpy, Btu/lb. The substance considered is designated by a subscript, e.g., a for air, p for products, f for fuel, λ for the λ mixture
 Δh = change of unit thermal enthalpy, Btu/lb. The temperature limits for the change are indicated as $t_a \Delta t_a$. The subscript \odot denotes an isentropic process change
 δh_L = difference between the two Δh_{\odot} magnitudes defined in Fig. 2, Btu/lb
 ΔH = thermal enthalpy change for the combustion chamber flow from inlet fuel and air to outlet products, Btu/lb of inlet air
 HHV_o = fuel "higher heating value," Btu/lb of fuel
 IC = "incomplete combustion chemical energy," Btu/lb of air
 k = specific heat ratio c_p/c_v
 LHV_o = fuel "lower heating value," Btu/lb of fuel
 P = gas pressure. Subscripts 1 and 2 refer to combustion-chamber inlet and discharge respectively, and subscript 3 to turbine exhaust
 ΔP = combustion-chamber pressure drop, $P_1 - P_2$
 P^* = ideal pressure ratio, inlet to combustion chamber divided by the turbine exhaust pressure, P_1/P_3
 Q_s = combustion-chamber setting loss, Btu/lb of air
 t = temperature, F. Subscripts: a refers to air, p to products, f to fuel, o to reference state
 V = gas volume, ft³/lb of air supplied to the combustion chamber
 W_{net} = net gas-turbine plant work, Btu/lb of turbine flow
 Δ, δ as a prefix denotes a difference.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 47-A-32.

- η_c = combustion-chamber combustion efficiency. Subscript 1 refers to Equation [1] definition, subscript 2 refers to Equation [2] definition.
 η_M = combustion-chamber mechanical efficiency.
 η_o = combustion-chamber over-all efficiency
 η_T = turbine-shaft isentropic efficiency
 λ = gas mixture defined as H_2O , CO_2 , etc., products of combustion minus the O_2 used on complete combustion, lb/lb of fuel
 a denotes air
 f denotes fuel
 i denotes ideal system
 o denotes reference state except for η_o
 p denotes products of the combustion chamber
 \odot denotes an isentropic process

INTRODUCTION

MANY different groups and organizations are working on the gas-turbine plant combustion-chamber development. Consequently it is highly desirable that some standard definitions of efficiency applying to combustion-chamber performance be established. While this paper makes specific recommendations in this regard, it is aimed primarily at stimulating discussion with a view to early standardization.

Any standard definition of efficiency of a system should satisfy the following requirements:

- 1 The definition should be "thermodynamically significant," that is, it should relate to the efficiency of conversion of one form of energy to another desired form; or, it should compare actual system performance to that of an "ideal system."
- 2 The 100 per cent magnitude should represent the "asymptote of performance" of any actual system, that is, "defined perfection" should be an approachable goal.
- 3 It should allow ready critical comparison of performance of different systems.
- 4 It should be easy to apply in cycle analyses.
- 5 It should be readily calculable from experimental measurements.

The function of the combustion chamber of a gas-turbine plant is to produce hot gas under pressure from the "internal combustion" of a fuel with compressed air. Qualitatively, performance may be judged by the following criteria:

- 1 The gas must be "clean" to avoid fouling and erosion of the combustion chamber, turbine, and regenerator.
- 2 The "heat leak" (or "setting loss") must be "small" for maximum plant efficiency.
- 3 The combustion must "closely approach" thermodynamic completeness for maximum plant efficiency.
- 4 The flow friction expenditure must be "small" for maximum plant efficiency and output.
- 5 The energy requirements for the auxiliaries must be "small" for maximum plant efficiency and output.

Aside from the first criterion the remainder all relate to plant efficiency and are susceptible to quantitative expression. It is the purpose of this paper to define three efficiency expressions,

one a "combustion efficiency" relating to criteria 2 and 3; the second a "mechanical efficiency" relating to criteria 4 and 5; and the last an "over-all efficiency" combining the first two expressions and certain cycle parameters.

The foregoing combustion efficiency will also be compared to a definition stemming from the orthodox concept of the steam-generating efficiency of a boiler,

COMBUSTION EFFICIENCY

The effect of heat leak from the combustion chamber and incomplete combustion may be expressed in terms of a combustion efficiency. This term is defined and discussed in the following development, which parallels that given in reference (1).²

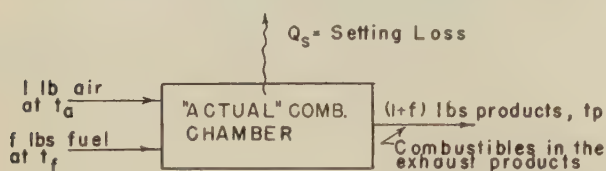


FIGURE 1a

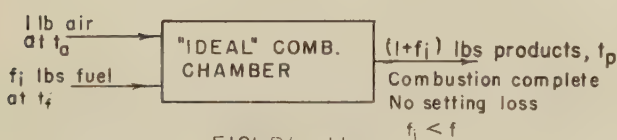


FIGURE 1b

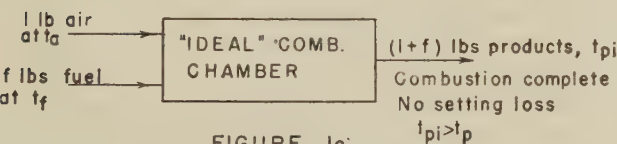


FIGURE 1c

FIG. 1 SCHEMATIC DESCRIPTIONS OF THE ACTUAL AND TWO IDEAL COMBUSTION-CHAMBER SYSTEMS
(Material quantities are given relative to one pound of air flow.)

Consider the schematic illustration of the "actual" combustion-chamber system shown in Fig. 1(a). Steady-state flow conditions are postulated so that for each pound mass of inlet air flow at temperature t_a there are f lb of fuel at t_f and $(1+f)$ lb of products at temperature t_p . The setting loss is symbolized by Q_s Btu/(lb of air). Any incomplete combustion is evidenced in the products by combustible components.

If it is granted that the function of the combustion chamber is to produce hot gas at a temperature t_p from the supply air at t_a with a minimum expenditure of fuel, then an ideal combustion chamber may be defined as depicted in Fig. 1(b). For this ideal system the following restrictions obtain: 1, Temperature of entering air, inlet fuel, and exit products are equal to the corresponding operating temperatures of the actual system; 2, combustion is adiabatic, that is, the setting loss is zero; and 3, the reaction of fuel and oxygen is "complete" so that no further combustion is possible for the exit products at the temperature t_p . As a result of these specifications the fuel requirement of the ideal system, f_i , will be less than that of the actual system, f .

The significance of the term "complete combustion" requires clarification. In most gas turbines using hydrocarbon fuels and air, the temperature t_p is sufficiently low so that dissociation is

negligible, and consequently, "thermodynamically complete" combustion is identical to "stoichiometrically complete" combustion. However, for combustion-chamber operation at higher temperatures (as, for instance, in ramjets), dissociation may be significant. In such cases, to secure true thermodynamic significance to the concept of the ideal combustion chamber, complete combustion should mean thermodynamically complete, as can be calculated from the equilibrium constants for the chemical reactions which form the combustion process.

The ideal combustion chamber, Fig. 1(b), evolved from the concept that the function of the combustion system was to produce products at temperature t_p from supply air at temperature t_a with a minimum expenditure of fuel. Another logical ideal combustion-chamber system would result from the concept that the function was the production of the maximum products temperature from the expenditure of the same amount of fuel, f , as for the actual system. Such an ideal combustion chamber is described in Fig. 1(c). For this system the following restrictions obtain: 1, Temperature of entering air and inlet fuel are equal to the corresponding temperatures of the actual system; 2, the fuel flow, f , is identical to that of the actual system; 3, combustion is adiabatic; and 4, the reaction of fuel and oxygen is complete. As a result of these specifications, the ideal system products temperature, t_{p_i} , and hence thermal enthalpy, h_{p_i} , will be higher than for the actual system, t_p and h_p .

A combustion efficiency may now be defined which will relate the actual combustion-chamber performance, Fig. 1(a), to the behavior of one of the ideal systems, Fig. 1(b) or 1(c). Thus the comparison of actual to minimum fuel requirement, Figs. 1(a) and 1(b) yields

$$\eta_{c1} = f_i/f \dots \dots \dots [1]$$

The comparison of actual to maximum products enthalpy (or temperature) yields, Figs. 1(a), 1(c)

$$\eta_{c2} = \Delta H_{\text{actual}} / \Delta H_{\text{ideal system 1c}} \dots \dots \dots [2]$$

where ΔH is the "thermal" enthalpy increase of the flow from the inlet air and fuel state to the exit products state. For the steady-flow systems considered, the thermal enthalpy function has the significance of internal thermal energy plus flow-work. The term "thermal" is used to qualify the enthalpy expression to indicate that chemical internal energy and the PV change due to chemical combination of the elements (and subsequent change in the number of molecules) are omitted. In Equation [2]

$$\Delta H_{\text{ideal system 1c}} = f \times HV.$$

the constant pressure "heating value" of f lb of the fuel at the reference state indicated by the subscript zero. This heating value is the decrease of chemical internal energy plus the decrease of PV due to the change in the number of molecules occurring during the combustion reaction. This PV change might be referred to as the "chemical" PV change to distinguish it from the thermal PV change associated with temperature.

It is necessary that the same reference state (pressure, temperature, and phase condition) be employed in evaluating the thermal enthalpy change, ΔH as used in the expression of HV . For instance, if the "higher heating value" HHV_o is employed the enthalpy of the water of combustion must be referred to a liquid state at temperature t_o , whereas if the "lower heating value", LHV_o , is used, the water enthalpy is referred to a vapor state at temperature t_o . Unlike the definition of steam-boiler efficiency, it is immaterial which heating value is employed as long as thermal enthalpies are referred to the proper reference state. Normally the reference state-pressure has a negligible influence in these calculations and may be omitted from further consideration. Equation [2] may now be written

² Numbers in parenthesis refer to the Bibliography at the end of the paper.

$$\eta_{c2} = \Delta H_{\text{actual}} / fHV_o \dots \dots \dots [3]$$

It is seen to be analogous to the conventional efficiency expression for a steam boiler with ΔH_{actual} comparable to the "useful enthalpy increase" from the feedwater state to the product steam state. For this reason Equation [3] has assumed some popularity in expressing test performance and for cycle calculations.

These calculations, in addition to the thermodynamic-properties data, require a specification of: 1, products composition; 2, products temperature; 3, inlet air temperature and humidity; 4, inlet fuel temperature; 5, HV_o of the fuel; 6, test fuel consumption per lb of air for the calculation of efficiency from test results; or 7, estimated combustion efficiency, η_{c2} , for the cycle prediction of fuel requirement.

In contrast, the defining Equation [1] for the combustion efficiency requires these data with the exception of the products composition and the addition of an ultimate analysis of the fuel. For normal hydrocarbon fuels a knowledge of the hydrogen/carbon ratio is sufficient.

Two combustion-efficiency expressions have been defined and the data required for their use have been specified. The selection of one definition as a standard in preference to the other depends on an evaluation of their relative merits in terms of the criteria of a suitable definition as expressed in the introduction. Such a comparison is presented in the following discussion.

Both definitions are thermodynamically significant in that they relate actual combustion-chamber performance to that of an "approachable" ideal system. However, η_{c2} , Equation [3], has the disadvantage in that some degree of arbitrariness is introduced by the necessity of specifying the reference state which enters into both the denominator $f \times HV_o$ term and the numerator enthalpy term. For any magnitude of $\eta_{c2} < 100$ per cent the selection of the reference state will influence the magnitude of the calculated result. This effect is small enough to be negligible in most calculations; nevertheless, it is desirable to eliminate such ambiguities. This may be accomplished by specifying arbitrarily the standard reference temperature, e.g., 60 F. However, for η_{c1} , Equation [1], no such specification is necessary. While HV_o enters into the calculation of the ideal fuel requirement, f_i , any reference state with the correct heating value will yield identical results.

Either definition is adequate for allowing critical comparison of the "combustion performance" of two or more combustion chambers, aside of course from the question of temperature uniformity of the exit products.

With respect to the criteria of ease of evaluation from test results or use in cycle calculations, η_{c1} , Equation [1], is superior to η_{c2} , Equation [3]. This may be demonstrated by writing an energy balance and by considering the details of evaluation of the terms in Equations [1] and [3] for a combustion chamber.

Refer to the actual combustion-chamber system described in Fig. 1(a). For steady-state flow conditions the following energy-balance expression results

$$h_{a,t_a} + fHV_o + f \times h_{f,t_f} = (1 + f)h_{p,t_p} + Q_s + IC \dots [4]$$

where Q_s is the setting loss and IC the incomplete-combustion loss. All thermal enthalpy terms h are referred to the reference state (defined by t_o and phase condition) as employed in the expression of HV_o . The temperature subscripts on the h terms refer to t_a , t_f , t_p as shown in the illustration, Fig. 1(a).

For the ideal system, Fig. 1(b), Equation [4], with Q_s and IC , zero, reduces to

$$h_{a,t_a} + f_i HV_o + f_i h_{f,t_f} = (1 + f_i)h_{p,t_p} \dots \dots \dots [5]$$

For the remainder of this discussion it will be assumed that the

inlet air is free of water vapor. While the effect of inlet water vapor may be taken into account by additional terms these will be omitted here for clarity. The products leaving the combustion chamber consist of the CO_2 , H_2O , and other components of complete combustion, plus excess air, plus N_2 associated with the oxygen used. After the method of references (2) and (3), it is convenient from the point of view of the calculations to consider the exhaust products as a two-component mixture, a dry-air component plus a " λ " component. The λ component is a "nonphysical" defined mixture consisting of the H_2O , CO_2 , and other products of combustion minus the oxygen consumed on combustion. On this basis, $(1 + f_i)$ lb of products consists of 1 lb of air plus f lb of component λ and thus

$$(1 + f_i) h_{p,t_p} = h_{a,t_p} + f_i h_{\lambda,t_p} \dots \dots \dots [6]$$

Introducing Equation [6] into Equation [4] and solving for the ideal fuel requirement results in

$$f_i = t_a \Delta t_p h_a / (HV_o + h_{f,t_f} - h_{\lambda,t_p}) \dots \dots \dots [7]$$

The numerator, $t_a \Delta t_p h_a$, the enthalpy change of air from inlet temperature t_a to a temperature equal to that of the products t_p , may be accurately evaluated from tables such as reference (4) or, more conveniently, from charts such as contained in reference (3). The heating value HV_o , either "higher" or "lower," must be specified (any reference state may be employed). The enthalpy of the fuel at temperature t_f , referred to t_o as a datum can be approximated from the specific-heat capacity for the fuel (0.50 Btu/(lb deg F) for most liquid hydrocarbons. High accuracy in this term is not necessary as it is usually very small relative to the HV_o term. The enthalpy of the λ component may be evaluated from the following information: 1, An ultimate analysis of the fuel, and 2, the equilibrium constants as a function of temperature t_p . For most hydrocarbon fuels, and for $t_p < 2500$ F, negligible error is introduced by the assumption of zero dissociation. Then, from the hydrogen/carbon ratio, the stoichiometric equations and available thermodynamic properties data for CO_2 , H_2O , and O_2 , it is possible to prepare a table or chart of the enthalpy of the λ component vs. temperature. Two such charts are available in reference (3), one for a hydrogen/carbon ratio of 0.15 corresponding to a Diesel-type oil and the other for 0.18 corresponding to gasoline or kerosene. These charts were prepared for use with the lower heating value LHV_o , that is, for a gaseous reference phase for the water of combustion. Consequently, if the HHV_o is given, it is either necessary to evaluate LHV_o from it or add to the h_{λ} the latent heat of vaporization of the water of combustion per lb of fuel at temperature t_o .

As an illustration of the use of Equation [7], consider the problem of determining the ideal fuel requirement, f_i , given the following:

Fuel data:

$HHV_o = 19,500$ Btu/lb for $t_o = 60$ F
 $t_f = 100$ F
 specific heat = 0.5 Btu/lb deg F
 hydrogen/carbon ratio = 0.15 (Diesel oil)
 fuel consumption $f = 0.00856$ lb/lb air
 Air inlet temperature $t_a = 650$ F
 Products exit temperature $t_p = 1200$ F

From the air charts of reference (3)

$$t_a \Delta t_p h_a = 650 \text{ F } \Delta_{1200 \text{ F}} h_a = 316.6 - 173.2 = 143.4 \text{ Btu/lb air}$$

For the given hydrogen carbon ratio, 1.17 lb of water of combustion per lb of fuel is formed. Thus using steam-table data for the latent heat of vaporization at 60 F

$$LHV_o = 19,500 - 1.17 \times 1060 = 18,260 \text{ Btu/lb of fuel}$$

The enthalpy of the liquid fuel referred to the reference temperature, $t_o = 60 \text{ F}$, is

$$h_{f,tf} = h_{f,100\text{F}} = (100 - 60) \times 0.5 = 20 \text{ Btu/lb of fuel}$$

(about 0.1 per cent of the LHV_o term)

The enthalpy of the λ component referred to the reference state temperature of 60 F is evaluated as an enthalpy difference from chart 11 of reference (3) as

$$h_{\lambda,t_p} = h_{\lambda,1200\text{F}} = 823 - 70 = 653 \text{ Btu/lb of fuel}$$

Then, from Equation [7]

$$f_i = 143.4 / (18,260 + 20 - 653) = 0.00814 \text{ lb/lb air}$$

The combustion efficiency from Equation [1] and the given $f = 0.00856 \text{ lb/lb}$ is then

$$\eta_{c1} = f_i / f = 0.00814 / 0.00856 = 95.0 \text{ per cent}$$

It is evident that the foregoing calculations are easily accomplished provided thermodynamic charts for the properties of the λ component are available. If they are not available, it is necessary to work with the CO_2 , H_2O , and "negative" O_2 components of the mixture using Dalton's law for the enthalpy of the mixture as the algebraic sum of the enthalpies of the components, duly weighted in accordance with their mass fraction expressed as lb/lb of fuel. It is to be emphasized again that the use of a negative oxygen component is for algebraic convenience only, and that such a negative component can have no physical existence.

Refer now to the ideal system Fig. 1(c). For this adiabatic and complete combustion system Equation [4] reduces to

$$h_{a,t_a} + f \times HV_o + fh_{f,tf} = (1 + f)h_{p,t_{pi}}$$

The increase in thermal enthalpy for this ideal system is

$$\Delta H \text{ ideal system } 1_c = (1 + f) h_{p,t_{pi}} - h_{a,t_a} - fh_{f,tf} = f \times HV_o \quad [8]$$

as previously stated. From Equation [4] the actual increase in thermal enthalpy is

$$\Delta H = \{(1 + f)h_{p,t_p} - h_{a,t_a} - fh_{f,tf}\} \dots \dots [9]$$

Then the combustion efficiency, in accordance with Equation [3], may be expressed

$$\eta_{c2} = \{(1 + f)h_{p,t_p} - h_{a,t_a} - fh_{f,tf}\} / f \times HV_o \dots \dots [10]$$

For the problem of calculating η_{c2} from experimental data, the composition of the products is required. Further, since f appears both in the numerator and denominator, the calculation for f for any given η_{c2} is more difficult relative to the calculation from η_{c1} . It is apparent therefore that in rigorous application, Equation [2] and the resulting computing Equation [10] are more difficult to apply than Equation [1]. A useful approximation can be made, however, which will simplify the use of Equation [10]. The actual products will have approximately the same thermal enthalpy-temperature relation as the products of complete combustion. Then, it becomes convenient to consider, as before, the actual products as consisting of the two-component air- λ mixture. On this basis Equation [9] may be written

$$\Delta H \cong t_a \Delta_{tp} h_a + f(h_{\lambda,t_p} - h_{f,tf}) \dots \dots [11]$$

and Equation [10] becomes

$$\eta_{c2} = \frac{t_a \Delta_{tp} h_a}{f \times HV_o} + \frac{h_{\lambda,t_p} - h_{f,tf}}{HV_o} \dots \dots [12]$$

Calculations from this equation can be simplified by the use of the charts of reference (3). As an illustration, consider the previously given numerical data. Using the charts of reference (3), as before, for the evaluation of $t_a \Delta_{tp} h_a$ and h_{λ,t_p} , Equation [12] yields

$$\eta_{c2} = \frac{143.4}{0.00856 \times 18,260} + \frac{653 - 20}{18,260} = 0.919 + 0.0347 = 95.4 \text{ per cent}$$

This compares to the previous result for η_{c1} of 95.0 per cent.

This lack of agreement between η_{c1} and η_{c2} is characteristic of the basic difference in the definitions, and ranges from zero per cent at $\eta_c = 100$ per cent to about 1.2 per cent at $\eta_c = 90$ per cent, reference (1). For this reason alone it is important from the point of view of standardization to settle on either one or the other of the definitions. It is proposed that the defining Equation [1] for η_{c1} be selected in preference to the defining Equation [2] for η_{c2} in view of 1, the lack of dependence of the magnitude of the combustion efficiency on the arbitrary selection of a reference state for the heating-value expression; 2, the more ready rigorous application in calculations; 3, the more direct association with the function of a combustion chamber to produce products at a specified outlet temperature, t_p ; and 4, the fact that exhaust-gas composition is not required for the calculation of efficiency from test data.

MECHANICAL AND OVER-ALL EFFICIENCY

A combustion chamber with a high η_c may be less desirable than a unit with a poorer combustion efficiency if the pressure drop flow friction expenditure is relatively large. As flow friction subtracts directly from the turbine work derivable from the products, its effect is reflected in both reduced plant thermal efficiency and rated output. Similarly, the energy requirements for auxiliaries such as fuel pumps, cooling air blowers, and the like, also reduce rated plant output and thermal efficiency.

A method of expressing these "mechanical inefficiencies" of the combustion chamber on an energy basis comparable to that of the previously considered "combustion inefficiencies" would allow a quantitative comparison of the thermodynamic performance of several units. The purpose of the following discussion is to define a "mechanical efficiency" term, η_M , descriptive of combustion-chamber performance, and to combine it with η_c and certain cycle parameters, to arrive at a thermodynamic or "over-all" combustion-chamber efficiency, η_o . While η_M , the mechanical efficiency, depends primarily on the combustion chamber in question, in order to express mechanical losses in terms of plant performance it is necessary to introduce certain gas-turbine cycle parameters. In effect then, the over-all efficiency, η_o , is not solely a combustion-chamber characteristic as it defines performance of the combustion chamber within a particular gas-turbine plant.

The combustion-chamber mechanical efficiency will be defined in terms of the relative performance of the actual and an ideal combustion chamber with respect to the isentropic work available at combustion-chamber discharge pressure. The ideal combustion chamber for this definition will be specified as follows:

- 1 The same products temperature and composition as the actual system.
- 2 The same inlet pressure but zero pressure drop in passage through the chamber. Note that on an exact basis "total pressure" should be used in place of static pressures.
- 3 Zero auxiliary energy requirement.

As a result of these specifications, the energy derivable from an

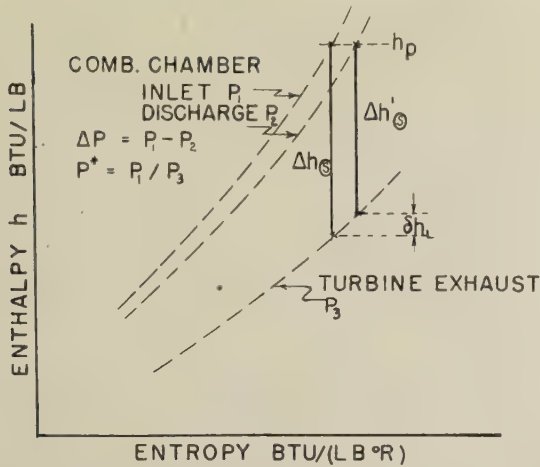


FIG. 2 ISENTROPIC TURBINE PROCESSES WITH AND WITHOUT COMBUSTION-CHAMBER PRESSURE DROP ILLUSTRATING THE LOSS OF TURBINE AVAILABLE ENERGY δh_L

isentropic turbine process is pictured as Δh_{\odot} in Fig. 2, where P_1 is the ideal combustion-system discharge pressure identical to the actual inlet pressure. In contrast, for the actual chamber, in view of the flow pressure drop $\Delta P = P_1 - P_2$, the reduced isentropic turbine available energy $\Delta h_{\odot}'$ results. Consequently, the isentropic energy loss chargeable to flow friction pressure drop is

$$\delta h_L = \Delta h_{\odot} - \Delta h_{\odot}' \quad [13]$$

All the indicated enthalpy changes are expressed per lb of turbine flow corresponding to combustion-chamber products flow.

In addition to this loss, the actual combustion chamber requires mechanical energy for the auxiliaries to the extent of E Btu/lb of turbine flow. The total mechanical energy loss is then $(\delta h_L + E)$ and, in accordance with the definition for mechanical efficiency

$$\eta_M = 1 - \frac{(\delta h_L + E)}{\Delta h_{\odot}} \quad [14]$$

Some objection exists here in giving the auxiliary energy E the same "value" per unit energy as δh_L . Since E is derived from the actual turbine output its value based on turbine inlet conditions would be E/η_T where η_T is the turbine efficiency. However, E is generally small relative to δh_L . Further, it may be argued that the combustion chamber should not be charged with actual turbine inefficiency. For these reasons the evaluation implied in Equation [14] is considered as justified.

The quantity Δh_{\odot} in Equation [14] may be evaluated from a knowledge of the products temperature leaving the combustion chamber and the "ideal pressure ratio," $P^* = P_1/P_3$. For a cycle with two turbines in series without an intermediate reheat stage, P_3 would correspond to the exhaust of the lower-pressure turbine.

The mechanical losses specified by η_M , Equation [14], are not directly comparable to the combustion losses specified by Equation [1]. This arises from the fact that actual gas-turbine cycles operate with turbine efficiencies less than 100 per cent (about 80–90 per cent) and with net work ratios (nwr) substantially less than unity (about 0.20 to 0.45). However, for any particular gas-turbine plant it is possible to evaluate combustion-chamber losses in terms of their influence on plant specific fuel consumption. From this viewpoint an over-all combustion-chamber efficiency may be defined as follows: η_o compares net work per lb of fuel of the gas-turbine plant and actual combustion

chamber, to the net work per lb of fuel from the plant modified to operate with an ideal combustion chamber having $\eta_M, \eta_c = 100$ per cent.

For the plant with the ideal combustion chamber there is an increase of net plant work to the extent of $\Delta h_{\odot} \eta_T (1 - \eta_M)$, and a reduction of fuel requirement from f to f_i . The slight difference in Δh_{\odot} due to ideal and actual products difference in composition may be neglected. Then, if W_{net} denotes the actual cycle net work (Btu/lb of turbine flow), the definition of η_o yields

$$\eta_o = \frac{W_{net}}{f/(1+f)} \bigg/ \frac{W_{net} + \Delta h_{\odot} \eta_T (1 - \eta_M)}{f_i/(1+f_i)}$$

The ratio $(1+f)/(1+f_i)$ in this equation closely approximates unity, and introducing $\eta_c = f_i/f$ the expression becomes

$$\eta_o = \eta_c \left[\frac{W_{net}}{W_{net} + \Delta h_{\odot} \eta_T (1 - \eta_M)} \right] \quad [15]$$

which may be rearranged to yield

$$\eta_o = \eta_c \bigg/ \left[1 + \frac{1 - \eta_M}{(W_{net}/\Delta h_{\odot} \eta_T)} \right] \quad [16]$$

It is seen that the plant parameters required, in addition to the η_c, η_M combustion-chamber characteristics, are turbine efficiency, η_T , and net cycle work relative to the isentropic available energy ($W_{net}/\Delta h_{\odot}$).

For the case of a plant with just a single combustion chamber, the term $W_{net}/\Delta h_{\odot} \eta_T$ has the approximate significance of the cycle "net work ratio," i.e., net work to turbine work ratio. It is seen then, from Equation [16], that mechanical inefficiency is multiplied by the reciprocal of the net work ratio in so far as the effect on plant cycle efficiency is concerned. As an example, for a net work ratio of $1/3$, a one per cent mechanical inefficiency has a 3 per cent effect on plant thermal efficiency. Furthermore, plant "size" for a specified rating must be increased to compensate for $(1 - \eta_M)$ in the same proportion as plant thermal efficiency is decreased.

Equation [16] will also serve for a particular combustion chamber in a reheat cycle. In this case η_T and Δh_{\odot} apply to the particular turbine between the combustion chamber in question and the next reheat chamber.

In experimental work the combustion-chamber friction performance is normally measured in terms of pressures, i.e., upstream pressure P_1 and pressure drop ΔP . If flow kinetic-energy changes are of a significant magnitude "total pressures" may be used in place of static pressure. Then, this data may be used for the evaluation of Δh_{\odot} and δh_L which enter into the calculation of η_M and η_o , Equations [14] and [16]. Methods and thermodynamic data for calculating δh_L and Δh_{\odot} are given in references (3) and (4).

Fig. 3 was prepared as an aid in these calculations. It may be shown (5) that the ratio $(\delta h_L/\Delta h_{\odot})/(\Delta P/P_1)$ is a function of the ideal expansion ratio, P^* , the fraction pressure drop, $\Delta P/P_1$, and $k = c_p/c_v$ of the turbine working substance. However, this ratio is not very sensitive to the fraction pressure drop for $\Delta P/P_1$ less than 10 per cent, and the influence of the gas temperature into the turbine is relatively small. Consequently, a plot of $(\delta h_L/\Delta h_{\odot})/(\Delta P/P_1)$ vs. the ideal expansion ratio, P^* , calculated for a given $\Delta P/P_1 = 0.05$ and turbine inlet temperature of 1800 R, will apply with good accuracy to a range of $\Delta P/P_1$ from 0–10 per cent and a range of turbine inlet temperatures from 1200 to 2200 R.

As an illustration of the use of Fig. 3 and the application of Equations [14] and [16], consider the problem of evaluating η_M

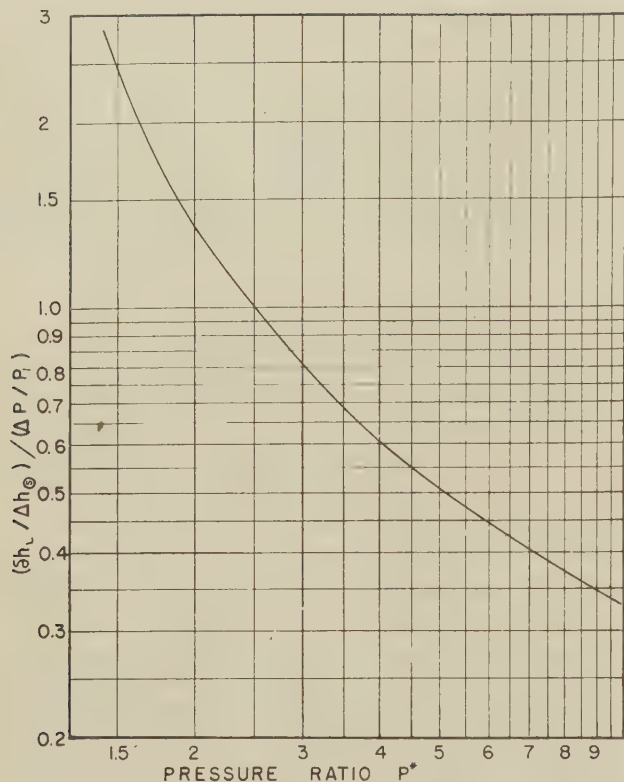


FIG. 3 THE RELATION BETWEEN COMBUSTION-CHAMBER PRESSURE DROP FRACTION, $\Delta P/P_1$, LOSS OF ISENTROPIC TURBINE WORK FRACTION, $\delta h_L/\Delta h_\odot$, AND THE IDEAL PRESSURE RATIO $P^* = P_1/P_2$ (Refer to Fig. 2 for a description of the turbine processes.)

and η_o given the following data for a "no-reheat" cycle

Combustion chamber:

$$\eta_c = 0.98$$

$$\Delta P/P_1 = 0.04$$

$$\text{auxiliary energy requirements} = 3 \text{ hp}/1000 \text{ shp}$$

Turbine:

$$\text{ideal expansion ratio } P^* = 4.00$$

(based on combustion-chamber inlet pressure)

$$\eta_T = 0.85$$

$$\text{inlet temperature} = 1800 \text{ R}$$

Plant:

$$\text{specific flow rate} = 50 \text{ lb/shp-hr}$$

For the indicated specific flow rate and auxiliary energy requirement

$$E = 3 \times 2545 / (1000 \times 50) = 0.153 \text{ Btu/lb of turbine flow}$$

For the given $\Delta P/P_1 = 4$ per cent, and $P^* = 4.00$, Fig. 3 yields

$$\left(\frac{\delta h_L}{\Delta h_\odot} \right) / \left(\frac{\Delta P}{P_1} \right) = 0.61$$

$$\text{so } \delta h_L / \Delta h_\odot = 0.61 \times 0.04 = 2.44 \text{ per cent}$$

Δh_\odot is evaluated from the charts of reference (3) as 143.9 Btu/lb (based on pure air properties as an approximation). Consequently, $E/\Delta h_\odot$ is only 0.106 per cent, a relatively small magnitude compared to $\delta h_L/\Delta h_\odot$. From the given specific flow rate and the foregoing evaluation of Δh_\odot

$$W_{\text{net}} / \Delta h_\odot = \frac{2545/50}{143.9} = 0.354$$

From Equation [14] and the foregoing results

$$\eta_M = 100 - (2.44 + 0.106) = .9745 \text{ per cent}$$

From Equation [16] and the foregoing results

$$\eta_o = 0.98 / \left[1 + \frac{0.0255}{0.345/0.85} \right] = 0.98/1.0628 = 92.2 \text{ per cent}$$

Note that the 2.55 per cent "mechanical loss" has a 6.3 per cent influence on plant efficiency, and 6.3 per cent of the plant size could properly be charged to the combustion-chamber losses. In contrast, it is to be emphasized that the combustion losses of 2 per cent have only a 2 per cent influence on thermal efficiency and no influence on plant size.

SUMMARY AND CONCLUSIONS

The foregoing discussion and analysis of combustion-chamber efficiency concepts may be summarized as follows:

1 In order to arrive at a significant definition of efficiency it is necessary to specify clearly the functions of a combustion chamber. Further, it is necessary that any definition be thermodynamically meaningful and be readily applicable in the expression of test performance and in cycle analysis.

2 It appears necessary to employ three distinct concepts of efficiency, one a combustion efficiency, η_c , expressive of incomplete-combustion and setting losses; the second, a mechanical efficiency, η_M , relating to the flow friction losses and auxiliary-drive mechanical energy requirements; and third, an over-all efficiency η_o , expressive of the effect of all of these losses on gas-turbine plant efficiency. The over-all efficiency will allow a critical comparison of several combustion chambers designed for a given plant.

3 The combustion efficiency is independent of the plant cycle and depends only on the combustion chamber in question. The mechanical efficiency is in some measure dependent on the cycle in that it is necessary to specify an "ideal expansion ratio", P^* , corresponding to the ratio of combustion-chamber inlet pressure and the exhaust pressure of the turbine following the combustion chamber.

4 The over-all efficiency, which expresses the thermodynamic performance of the combustion chamber within the gas-turbine plant cycle, is a function of the combustion- and mechanical-efficiency terms, and in addition, the cycle parameter of the ratio of net work of the cycle to the "downstream turbine" work derivable from the ideal expansion ratio, P^* .

5 All these efficiency terms compare performance of, or with, the actual combustion chamber to the performance of, or with, an "ideal combustion chamber" which is capable of exact definition.

It is recommended that consideration be given to these efficiency concepts with a view to eventually deciding on standard definitions so that reported test results may be analyzed and cycle analyses interpreted without the ambiguities which currently exist.

ACKNOWLEDGMENT

Mr. A. Amorosi, of the U. S. Navy Bureau of Ships, and Mr. A. T. Scott, of the I.T.E. Circuit Breaker Co., contributed materially to the concepts presented here. Mr. Martin A. Mayers, of the W. M. Kellogg Company, first stressed the importance of establishing standard definitions for combustion-chamber combustion efficiency by pointing out the discrepancy resulting from the use of two current definitions.

BIBLIOGRAPHY

- 1 "Gas-Turbine Plant Combustion Chamber—Concept of Efficiency," by A. L. London, Bureau of Ships Research Memorandum No. 7-44, August, 1944.

2 "A Thermodynamic Investigation of Gas-Turbine Cycles," by R. V. Kleinschmidt, Bureau of Ships Research Memorandum No. 1-42, November, 1942.

3 "Gas-Turbine Gas Charts," by A. Amorosi, Bureau of Ships Research Memorandum No. 6-44, December, 1944. (Available from the Government Printing Office.)

4 "A Table of Thermodynamic Properties of Air," by J. H. Keenan and J. Kaye, John Wiley & Sons, Inc., New York, N. Y., 1944.

5 "Gas-Turbine Plant—Evaluation of Flow Friction Mechanical Energy Losses," by A. L. London, Bureau of Ships Research Memorandum No. 8-44, December, 1944.

Discussion

G. M. DUSINBERRE.³ Combustion efficiency is a natural concept, and the author has done well in directing attention to alternative viewpoints as to its precise definition.

But after that, Voltaire's saying comes to mind: "*Si Dieu n'existait pas, il faudrait l'inventer.*" If an "efficiency" does not exist, is it really necessary to invent one?

In the steam cycle, pressure drop in the superheater is quite analogous to combustion-chamber pressure drop. But we have not found it helpful to talk about "superheater mechanical efficiency." By far the greatest thermodynamic loss in the combustion chamber is that due to the mixing of excess air. This dilution is necessary for turbine life. However, the writer would not suggest setting up a "turbine metallurgical efficiency."

In fact, we might well add to the author's criteria the following: "An efficiency, to be ascribed to a particular system, must be calculable from measurements taken on that system, plus, at most, the ambient temperature and pressure."

The conclusion is that certain performance factors can be treated, with more logic, simply as losses rather than to force them, Procrustean fashion, into an arbitrary efficiency.

Aside from the mixing loss noted, what are the available energy losses chargeable to the combustion-chamber pressure drop? Referring to Fig. 4 of this discussion, let A be the actual end point

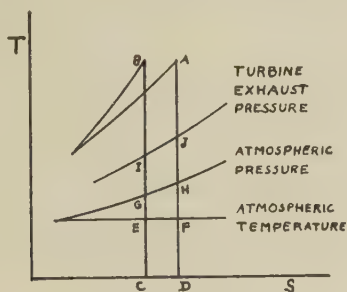


FIG. 4

of a process, and let B be the end point of a process performed from the same initial point in some prescribed reversible fashion so as to "correspond" to the actual process.

(a) Then the minimum loss chargeable to the pressure drop is the area $CEFD$. This cannot be recovered by any degree of thermodynamic perfection in subsequent apparatus.

(b) If we wish to accept the limitation of the constant-pressure lower line of the Brayton cycle, then we could charge the process with the loss $CGHDC$. This is unrealistic in ignoring the possibility of regeneration. So far we have involved only the actual process, the ideal process taken as "corresponding," and the ambient temperature, and pressure.

(c) If we continue to ignore regeneration and are willing to involve the working conditions of an extraneous system, we can

charge the combustion chamber with the loss $CIJDC$. This is the loss upon which the author's "mechanical efficiency" is based.

(d) It is not possible to show in any simple way the loss corresponding to the author's "over-all" efficiency. This efficiency, ascribed to the combustion chamber, involves not only the working conditions but also the actual performance of two systems extraneous to the combustion chamber itself.

So we have a choice of four losses which can be charged to combustion-chamber pressure drop. It seems to the writer that (a) is simple and thermodynamically fundamental. It can be evaluated readily in terms of the availability function⁴ ($b = h - T_0 s$).

If we wish to be more "practical," we can use (b). Then if we charge this loss to the combustion chamber in all cases, we can assign the regenerator an available energy credit. Otherwise it would be charged with a loss due to falling short of 100 per cent effectiveness.

The writer can see little in favor of (c) and less in favor of (d).

L. S. ECHOLS.⁵ The author has treated with considerable skill the problem of defining the performance relations of a combustion chamber to the unit in which it is to be used. However, the writer feels that this is not quite the problem as set forth in the title of the paper. Instead of defining a combustion efficiency evidently he has set forth a number of ways of defining a figure of merit for a combustion chamber. This is a worthwhile objective, but it should not become confused with combustion efficiency, as such.

Combustion efficiency is a term deeply rooted in our technical language, and it has a relatively simple meaning. To a chemist and to a thermodynamicist it is the degree to which the heat available in the fuel has been released, independently of reheat factors introduced by pressure loss and turbulence degeneration. Only a knowledge of the heat evolved in completing the combustion, or equivalent knowledge, can eliminate the effect of these reheat factors. This calls for a simple procedure for accomplishing this objective, and definite progress has been made along this line in England as described by Dr. Lloyd.⁶

It is a practical consequence of this approach that methods which measure the total heat developed in a process and attempt to correct for extraneous effects may, in the limit, be crude in comparison to methods which measure the residual undeveloped heat.

Consideration must be given to the problem of relating figures of merit to a broadly acceptable base-line definition of combustion efficiency, or, as a consequence, we may find ourselves embroiled in an endless argument as to the validity or acceptability of figures of merit for application to a wide variety of gas-turbine cycles. In the writer's opinion, we must start with the simple definition and relate clearly all derived quantities to it as a base line, if we are to secure agreement of all parties concerned.

Regarding the exclusive use of temperature rise as a measured quantity, it must be pointed out: (a) That an average exhaust temperature is implied and this requires not only a large number of temperature measurements but also the relative mass flows

⁴ If it is admissible to use a standard temperature, constant- b lines can be readily plotted on an $h-s$ chart as follows: Through the point of standard atmospheric pressure and temperature, rule a straight line tangent to the constant-pressure line at this point. Rule other constant- b lines parallel to this. Read b intercepts on the vertical enthalpy scale. We of course are free to introduce any arbitrary additive constant in the values of b itself.

⁵ Research Laboratories, Shell Oil Company, Inc., Wood River, Ill.

⁶ Refer to paper by Dr. Lloyd, page 335, of this issue of Transactions.

³ Division of Mechanical Engineering, University of Delaware, Newark, Del. Mem. ASME.

at each point of measurement. This is not easy. (b) We are approaching a temperature range wherein much doubt can be cast upon the existence of a measurable temperature in the thermodynamic sense. Evidence is accumulating to support the view that "temperature" is a very uncertain and almost indefinable, quantity above 2000 F. Engineers are struggling with inconsistencies of the order of 100 F among various methods of measurement in a regime where we are beginning to realize the simultaneous existence of different temperatures for different degrees of freedom of molecular motion, and at Mach numbers which produce serious errors in thermocouple measurements. These facts cannot be neglected in a practical choice of a method of measuring combustion efficiency or in rating combustion chambers. We should consider critically a variety of possible methods before approaching standardization. Specifically, it is suggested to include exhaust-gas analysis methods, both in complete and in simplified form.

J. P. LONGWELL.⁷ In view of the widespread use that apparently will be made of high-capacity combustors, such as in the gas turbine and the ramjet, it is felt that a standardized system of rating combustor performance will be of considerable value. The proposed definition of combustion efficiency based on the ratio of the quantity of fuel theoretically required to give the observed flame temperature to the amount actually used appears to offer definite advantage over other definitions and is at present used by a number of groups working on gas-turbine and ramjet combustors.

As pointed out by the author, losses of stagnation pressure which occur in the combustor also have an important effect on the over-all efficiency of the power plant. Agreement on a quantity used to characterize these losses may be considerably more difficult to arrive at, since in addition to the mechanical efficiency, suggested by the author, the ratio of stagnation pressure to total combustor inlet pressure, and a drag coefficient based upon combustor inlet conditions are also commonly used. These last two numbers are not efficiencies but can be easily used both for comparative purposes and for cycle evaluation.

In addition to the mechanical and chemical losses, the combustion itself causes a loss in stagnation pressure in the combustor. This loss in stagnation pressure corresponds to an increase in entropy in the same manner as mechanical friction losses and can be lumped with the friction losses for further use in power-plant evaluation. Table 1 of this discussion shows the ratio of stagnation pressure loss, caused by heat addition to a gas flowing through a tube of constant cross section, to the stagnation pressure at the combustor inlet. This calculation taken from NACA Technical Note 1180 assumed that the stagnation temperature was increased by a factor of 3.

TABLE 1 LOSS IN STAGNATION PRESSURE AS A FUNCTION OF COMBUSTION-CHAMBER INLET VELOCITY

Velocity before heating,	
fps	$\Delta P/P$
50	0.003
100	0.011
150	0.025
200	0.046
250	0.076

Since the loss depends on the velocity in the combustion chamber and therefore on combustion-chamber design, it could be charged to the combustion chamber and is a consequence of having a low frontal area. This loss, due to heating the stream, is frequently of approximately the same magnitude as the loss due to friction in aviation gas turbines and therefore is of equal importance. While it might not be desirable to lump the

mechanical friction loss and the loss due to heating, it would seem desirable to express both the same way.

A. AMOROSI.⁸ Quite naturally, we wish to evaluate gas-turbine combustion-chamber performance. One basis is efficiency. The very fact that combustion efficiency cannot be determined accurately has led to indifference as to the proper basis for comparison. However, because there is an instrumentation difficulty in determining combustion efficiency, there is all the more reason for establishing a standard so that secondary inconsistencies can be avoided. As for mechanical losses, it appears that our former practice of considering only combustion efficiency in quoting boiler efficiency has led us into a bad habit. The fact that mechanical losses should be included in the "heat-source" efficiency is strikingly apparent in the design of nuclear power plants where the pressure drop in the fluid being heated has a marked bearing upon the design of the nuclear reactor. If it is improperly taken into account, erroneous conclusions can be obtained. The fact remains that before we can evaluate combustion chambers properly, a rational basis for determining combustion-chamber efficiency should be established. Readers who consider carefully the points brought out by the author, will agree with his recommendations both as to the method of representing combustion efficiency and over-all combustion-chamber efficiency.

One might wonder why the author has chosen his over-all combustion chamber efficiency η_0 , based on its effect on net work of the over-all gas-turbine plant, instead of on a basis similar to turbine or compressor efficiency. Since η_c takes into account a thermal (or chemical) energy conversion and η_m a mechanical energy conversion, a moment's reflection will indicate that it would be difficult to combine the two into one efficiency other than on the basis of their effect on net work or net efficiency of the plant. Furthermore, it is customary to evaluate auxiliary equipments on their effect on net work or net efficiency. Therefore his choice seems to be well justified.

The acceptance of this method of determining combustion efficiency is a good first step in evaluating combustion chambers. Another valuable basis for evaluating combustion chambers is the ratio of maximum to average temperature rise through the chamber. The ability of a turbine to convert thermal energy to mechanical energy is a function of the average inlet temperature to the turbine, whereas the mechanical design problem is to a great degree influenced by the maximum gas temperature. Therefore the maximum to average temperature rise through the combustion chamber should also be reported as a measure of combustion-chamber performance.

J. K. SALISBURY.⁹ The author has chosen to evaluate the thermodynamic losses or mechanical efficiency in a combustion chamber by using the decrease in turbine available energy as the chief factor. In considering pressure drop through the combustion chamber, or for that matter through any portion of a gas-turbine cycle, two methods are possible. One may consider, as the author has done, that pressure drop lowers the turbine available energy or, alternatively, that it increases the compressor pressure ratio.

The writer chooses to use the latter method in an analysis presented as a discussion of a recent paper¹⁰ because it greatly

⁸ Mechanical Engineer, Bureau of Ships, Navy Department, Washington, D. C. Mem. ASME.

⁹ General Engineering and Consulting Laboratory, General Electric Company, Schenectady, N. Y. Mem. ASME.

¹⁰ "Some Effects of Pressure Loss on the Open Cycle Gas-Turbine Power Plant," by J. I. Yellott and E. F. Lype, Trans. ASME, vol. 69, 1947, pp. 903-911; discussion by J. K. Salisbury in Trans. ASME, vol. 70, February, 1948, pp. 139-141.

⁷ Esso Laboratories, Elizabeth, N. J.

simplifies the analysis. If pressure drops are considered to reduce the turbine pressure ratio, it is implicit that the turbine-nozzle area will have to be increased to pass the mass flow of the compressor at a given constant speed. Thus the analysis is invalid for a physical gas turbine. If the pressure drop is considered to increase the compressor pressure ratio in an axial-flow gas-turbine plant, there is a negligible change in mass flow due to the intrinsic constant-flow characteristic of this type of compressor. For this reason, additional pressure drops introduced into the system after fabrication of the component machines actually will increase the compressor pressure ratio. This alone lends logic to the viewpoint.

If the turbine-inlet pressure is considered to be reduced by the introduction of pressure drop, it becomes necessary in calculation to take into account deviations from constant specific heat normally present even for air at elevated temperatures, as well as the effect of combustion products on specific heat. Additional uncertainty as to the true effect of pressure drop is introduced by the question as to whether this rapid change in specific heat at the higher temperature levels may be partially responsible for the calculated loss. If the compressor pressure ratio is considered to be increased by a pressure drop no attention need be given to deviations of specific heat from normally used values because of their constancy at normal compressor pressures.

Even in the case of a reheat-gas-turbine cycle, this method may be used. In this instance, when a pressure drop occurs in the reheat combustion chamber, an increase in the high-pressure turbine pressure results, thus destroying the similarity to actual physical conditions which existed for the simple cycle, by necessitating a decrease in high-pressure-turbine nozzle area for the same mass flow. Pressure drops in the high-pressure combustion chamber may be added to those in the reheat combustion chamber, and the two reflected in an increase in compressor pressure.

For a simple cycle, pressure drops in the exhaust of the turbine are analogous to those in the reheat combustion chamber, and, by the same token, such pressure drops may also be reflected in the compressor pressure ratio.

One must be cautious in applying either of the two alternative methods. If a cycle is chosen such that the operating conditions are not on the flat part of the thermal-efficiency versus cycle-pressure-ratio curve, a change in pressure ratio of either turbine or compressor will include part of the increment or decrement in efficiency delineated by the cycle-efficiency curve, and will not represent the true pressure-drop effect. This fact is well brought out in the exposition by Yellott and Lyne.¹¹

A. H. SHAPIRO.¹² To the list of desirable requirements in this paper should be added the specification that the definition of efficiency be such that its magnitude depend on as few other parameters as possible.

In addition, it might be argued that the definition should be "functionally significant," rather than "thermodynamically significant," that is, we must consider the object of the combustion process. In the gas-turbine plant, the object is to produce hot gases for the production of work in a turbine, and ultimately for the production of net power from the plant. It seems reasonable therefore to define the efficiency so as to relate clearly the performance of the combustion chamber with the net power and with the fuel consumption. In a steam boiler plant, the object is to produce hot gases for the transfer of heat to boiling water.

This suggests that the efficiency of this case relate the fuel consumption to the amount of heat available for heat transfer. To go farther afield, we might imagine that the combustion chamber had as its purpose to produce certain chemicals among the products of reaction, in which case the efficiency would be measured by the amount of desirable chemicals produced per unit fuel consumption.

On the basis of these considerations, it seems that the quantity denoted by η_{c1} is significant for gas-turbine combustors, while η_{c2} is significant for combustors of the boiler-plant type.

The question of defining a mechanical efficiency is complicated by the close interrelationships between the various components of the plant. The many diverse points of view brought out in the discussion following presentation of the paper suggest that no one definition of wide acceptance may be found, and that such a simple parameter as the fractional loss in stagnation pressure may in the end be of greatest service. This parameter at least has the virtue that, for a fixed combustor design with fixed entry conditions and fuel rate, it is independent of the arrangement of other equipment in the plant and of the pressure ratios employed; whereas the more complicated definitions of mechanical efficiency do involve a dependence on these latter factors.

Finally, the writer wishes to raise the question of whether it is possible to define a single parameter, such as an "over-all efficiency," for appraising the merits of combustor designs. Because of the many independent considerations which usually enter into the design of engineering equipment, the arbitrary combination of several parameters into one often obscures rather than clarifies the meaning of the parameter. In more concrete terms, a combustor of a certain over-all efficiency might prove relatively effective in one type of gas-turbine plant and relatively ineffective in a different type. By retaining the separate concepts of "combustion efficiency" and of fractional loss in stagnation pressure, this difficulty is avoided, and the merits of the combustor for each particular type of plant can be determined on an individual basis. This point of view is even more significant when considerations of frontal area, or of the economic aspects, play an important part in the design procedure.

D. H. FAX.¹³ The first criterion to be applied to an efficiency definition should be that of generality over the entire scale from zero to unity. The usual efficiency concept, the ratio of the actual energy conversion to that attained in an ideal process, satisfies this requirement.

The first combustion-efficiency definition proposed by the author is the ratio of two fuel requirements. This definition admits of ease of calculation but it breaks down at the zero end of the scale. A reasonable definition for the zero of combustion efficiency would be that attained in a properly adjusted Junkers calorimeter, i.e., all of the heat value of the fuel would go into setting loss, and the products would emerge at a temperature t' , the same as that obtained by mixing the air at t_a and the fuel at t_f without combustion. An examination of Equation [7] of the paper shows that, in this case, η_{c1} would be negative or positive depending upon whether t_a is greater or less than t_f .

The definition of η_{c2} conforms more nearly with the requirements stated in the first paragraph, but it suffers from the fact that its magnitude depends upon the arbitrary selection of a reference temperature t_0 . This difficulty can be traced to Equations [8] and [9] of the paper. Those equations would represent the respective enthalpy increases in an ideal and in an actual

¹¹ "Some Effects of Pressure Loss on the Open-Cycle Gas-Turbine Power Plant," by J. I. Yellott and E. F. Lyne, Trans. ASME, vol. 69, no. 8, 1947, p. 903.

¹² Associate Professor of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, Mass. Mem. ASME.

¹³ Assistant Professor of Mechanical Engineering, The Johns Hopkins University, Baltimore, Md.; and Consultant, Gas Turbine Project, U. S. N. Engineering Experiment Station, Annapolis, Md. Jun. ASME.

system only if the individual values of h were counted above absolute zero, but not if counted from t_0 as done by the author.

The writer would propose a combustion efficiency, η_{cs} , which he believes removes the objections encountered in the foregoing. Instead of Equation [4], write the energy equation as does Lichty¹⁴

$$h_{a,t_0} + f h_{f,t'} + f C = (1 + f) h_{p,t_p} + Q_s + IC \dots [17]$$

where the nomenclature is that of the author except that the enthalpies are counted above absolute zero and C represents the chemical energy of the fuel. One may also write

$$h_{a,t_0} + f h_{f,t'} = h_{a,t'} + f h_{f,t'} \dots [18]$$

where t' is the temperature resulting from mixing the air and the fuel without combustion.

The heat value of the fuel is defined as the energy released when the products are cooled to the initial temperature of the air-fuel mixture, that is

$$h_{a,t'} + h_{f,t'} + f C = (1 + f) h_{p,t'} + f HV_t' \dots [19]$$

Subtracting Equation [19] from [17] and introducing Equation [18]

$$(1 + f) t' \Delta_{tp} h_p + Q_s + IC = f HV_t' \dots [20]$$

From this one may write

$$\text{actual } \Delta H = (1 + f) t' \Delta_{tp} h_p$$

$$\text{ideal } \Delta H = (1 + f) t' \Delta_{tp} h_p = f HV_t'$$

and

$$\eta_{cs} = (1 + f) t' \Delta_{tp} h_p / f HV_t' \dots [21]$$

Equation [21] is entirely general and appears to the writer to be rigorously correct; however, it does not admit of ready calculation. Therefore, introducing the author's approximation that leads to Equation [11], the numerator of Equation [21] may be written

$$t' \Delta_{tp} h_a + f t' \Delta_{tp} h_{f\lambda}$$

To find HV_t' from the heat value known at some reference temperature t_0 , rewrite Equation [19], replacing the t' subscripts with t_0 . Subtracting the resulting equation from Equation [19]

$$f HV_t' = f HV_0 + t_0 \Delta_{tp} h_a + f t_0 \Delta_{tp} h_{f\lambda} - (1 + f) t_0 \Delta_{tp} h_p$$

Using the same approximation as in the foregoing equation, this reduces to

$$HV_t' = HV_0 + t_0 \Delta_{tp} h_{f\lambda} - t_0 \Delta_{tp} h_{p\lambda} \dots [22]$$

The final expression for η_{cs} becomes

$$\eta_{cs} = \frac{t' \Delta_{tp} h_a + f t' \Delta_{tp} h_{f\lambda}}{f [HV_0 + t_0 \Delta_{tp} h_{f\lambda} - t_0 \Delta_{tp} h_{p\lambda}]} \dots [23]$$

From Equation [18]

$$t' = (c_a t_a + f c_f t_f) / (c_a + f c_f)$$

where c_f is the fuel specific heat and c_a is the constant pressure specific heat of the air.

Equation [23] is not as formidable as it may appear. Using the data given in the author's illustrative example, $t' = 641$ F, $LHV_t' = 18,254$ Btu per lb fuel, and $\eta_{cs} = 0.9521$. This latter value compares with $\eta_{cl} = 0.950$ and $\eta_{ca} = 0.9521$ for the same data.

In this connection it should be pointed out that there is a

¹⁴ "Thermodynamics," by L. C. Lichty, McGraw-Hill Book Company, New York, N. Y. 1936, pp. 150-152.

numerical error in the author's evaluation of $\eta_{ca} = 0.945$. In fact, it may be shown that for all usual values of the variables, η_{ca} is always greater than η_{cl} , rather than the reverse as might be implied from the paper. Furthermore, η_{ca} will be greater than η_{cs} if the specific heat of the λ component is greater than that of the fuel, and less than η_{cs} if the reverse is true. With the data of the author's example, $\eta_{ca} > \eta_{cs}$, but only in the fifth significant figure, because the two specific heats referred to are practically the same.

A casual inspection of the foregoing results might lead one to feel that, since they are so nearly equal (particularly when compared with the accuracy of the data), the only basis on which to choose between them is that of ease of calculation. The writer cannot join in such an opinion. While the differences between the three efficiencies are in the present instance minuscule, given other data they might not be. A definition that is valid only over a restricted range is of questionable value because definitions tend, with use, to become separated from the restrictions implicit in them. An efficiency definition should be valid over as wide a range of the variables as possible and should be independent of reference conditions which are completely arbitrary, such as t_0 in this case. Of course in any given application, approximations are always permissible so long as they are within the accuracy of the data.

The mechanical and over-all efficiencies defined in the paper are useful concepts. However, it is also desirable from several points of view (certainly that of the designer) to define a parameter for mechanical losses in the combustion chamber that will be a function of the chamber alone and independent of the pressure ratio of the cycle in which it may be used. Such a parameter is the "energy loss" in the combustion chamber, $v \Delta p / J$, where v is the specific volume at the combustion chamber discharge, and Δp the drop in total pressure through the chamber. For small pressure drops, $v \Delta p / J$ may be replaced by δh_1 , Fig. 5 of this discussion. The concept of δh is more useful than that of

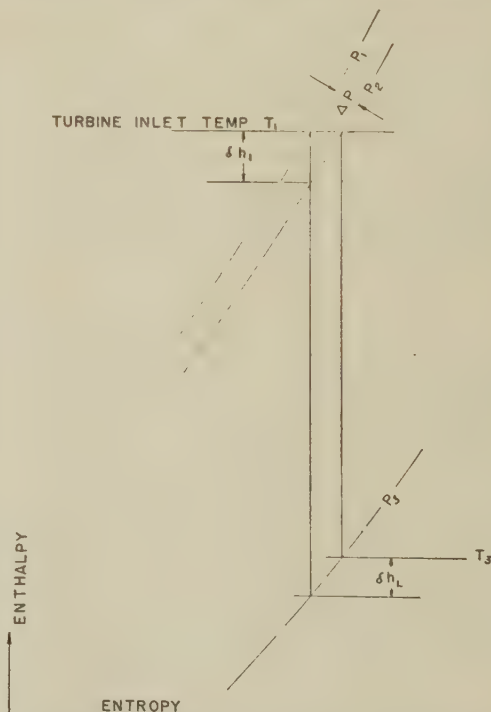


FIG. 5 RELATIONSHIP BETWEEN COMBUSTION-CHAMBER ENERGY LOSS δh_1 AND TURBINE AVAILABLE ENERGY LOSS δh_L
(Nomenclature is similar to that of author's Fig. 4.)

fractional pressure drop $\Delta p/p$ for several reasons. For one, it is easier to deal with when considering the effects of parasitic losses at part loads. From the Fanning friction formula it can be seen that, for isothermal flow, $\Delta p/p$ will vary with at least two variables, fluid velocity and temperature, whereas δh will vary with velocity alone (assuming insensitivity of friction factor with temperature). Going further, one might use the parameter $\delta h \div V^2/(2g_c J)$, which for isothermal flow would be fairly constant over a relatively wide range of velocity and temperature conditions.

Of course for a combustion chamber, $\delta h \div V^2/(2g_c J)$ will not be constant with load but will vary with one other parameter, generally taken as the ratio of absolute temperatures at the inlet and outlet of the combustion chamber. In any case, the use of $\delta h \div V^2/(2g_c J)$ requires two less parameters for correlation than does the use of $\Delta p/p$.

The concept of energy loss (rather than pressure drop) has been used before,^{15,16,17} and it should be furthered. The relationship between the two parameters is simply $\delta h_1 = RT_1 \Delta p/p$.

It is also easy to find the effect of combustion-chamber energy loss on the turbine available energy in a given cycle. Since

$$\left(\frac{\partial h}{\partial s}\right)_P = T, \text{ (see, e.g., Lichty, }^{18}\text{) one may write simply}$$

$$\frac{\delta h_L}{\delta h_1} = \frac{T_3}{T_1} = \left(\frac{P_3}{P_2}\right)^{\frac{k-1}{k}} \cong P^* \left(-\frac{k-1}{k}\right)$$

AUTHOR'S CLOSURE

The primary purpose of this paper, to stimulate discussion on the subject of gas-turbine plant combustion-chamber efficiency, has apparently been attained as evidenced by the thoughtful discussions of the reviewers. It is also evident from the discussions that early standardization is highly desirable before many more diverse opinions and definitions come into use.

Professor Dusenberre points out that combustion efficiency is a "natural" concept but implies that mechanical efficiency and over-all efficiency as defined by the author are not natural concepts and concludes therefore that there is no reason to "invent" them. To the author the comparison of "actual system performance" to that of an "ideal system" is a natural thermodynamic concept. All the efficiency terms presented in the paper are defined in this manner. Professor Dusenberre prefers to employ a mechanical energy "loss concept" in preference to an efficiency concept. This loss is evaluated by comparing actual system performance to ideal system performance. Three alternative methods of expressing performance are considered including the method employed as a basis for the author's η_M . The major purpose for "inventing" η_M was to provide a basis of comparison of "mechanical losses" and "combustion losses" in terms of the over-all efficiency, η_0 , or over-all loss of plant specific fuel consumption. Professor Dusenberre's availability function loss will not allow this objective. The isentropic available energy neglecting turbine exhaust pressure drop, while more realistic in this regard, is not quite as suitable as the recommended procedure of using the isentropic available energy based on the actual turbine exhaust pressure.

The author grants that Equation [16], a computing equation for η_0 , does not allow or credit the combustion chamber with the

greater possibility for regenerative heat transfer arising from the mechanical loss $(1 - \eta_M)$. While it would be desirable to include this effect, the resulting complexity of the equation would not make it worth while, especially so in view of the fact that even with high regenerator effectiveness (75 per cent), and plant thermal efficiency (25 per cent), the magnitude of this gain is in the order of less than 20 per cent of the loss, $(1 - \eta_0)$. For smaller amounts of regeneration and lower thermal efficiencies the effect is even less. Further, if η_0 , η_M , and η_c are known together with regenerator effectiveness and plant thermal efficiency, the regenerator effect can be readily estimated for any particular cycle. Note too that the recommended word definition of η_0 does not obviate the inclusion of the "regenerator recovery" by suitable modification of Equation [16].

Mr. Echols feels that any discussion of gas-turbine plant-combustion efficiency should be restricted to a consideration of "chemical reaction efficiency" only. To the author this appears as an unnecessary restriction, as certainly the engineer and thermodynamicist, as distinct from the chemist, must evaluate all significant losses. In the expression of internal-combustion-engine performance, for instance, it is not unusual to employ as many as four distinct efficiency expressions of performance. It is of course essential that any efficiency term be properly qualified if it is to be correctly interpreted.

Dr. Longwell points out that agreement on the quantity to be used to characterize flow-friction losses may be more difficult to arrive at than in the case of incomplete combustion and setting losses. He also indicates that there is an inherent mechanical loss, in addition to flow friction, as a result of the increase of gas temperature during combustion. This loss becomes quite significant for velocities before combustion of greater than 50 ft per sec. It seems to the author that this loss is properly chargeable as a mechanical energy loss and should be included in η_M .

The advantage of using η_M as proposed, instead of a per cent pressure drop or a combustion drag coefficient to characterize mechanical energy losses, is that it combines in a relatively simple manner with the combustion efficiency η_c and allows a comparison of the magnitudes of the component losses on a common basis, namely, their influence on plant specific fuel consumption.

It is evident that there is no unique common basis of evaluation and comparison of the component losses. Mr. Salisbury very ably points out the convenience in cycle analysis of using a "compressor work" evaluation of mechanical losses. Note, however, that the word definition of η_M as recommended by the author does not obviate the method of analysis proposed by Mr. Salisbury. Given η_M on a turbine basis it is a relatively simple matter to evaluate fraction pressure drop $\Delta P/P$ (see Fig. 3) and impose this as an additional pressure-ratio requirement on the compressor.

Should it be desirable to express η_M on a compressor work basis, by an equation similar to Equation [14], then a modification of Equation 16 may be used for an over-all efficiency.

$$\eta'_0 = \eta_c / \left[1 + \frac{1 - \eta'_M}{W_{net}/(\Delta h_{\odot}'/\eta_{comp})} \right] \quad [16a]$$

where the ($'$) notation denotes a compressor work basis. The term

$$\frac{1 - \eta'_M}{W_{net}/(\Delta h_{\odot}'/\eta_{comp})}$$

approximates the mechanical loss evaluation in terms of the plant specific fuel consumption. For the conditions of the numerical example of the paper it has a magnitude of 6.8 per cent (for a compressor isentropic efficiency of 85 per cent). The similar term of Equation [16]

¹⁵ "The Gas Turbine as a Possible Marine Prime Mover," by C. R. Soderberg and R. B. Smith, Trans. Society of Naval Architects and Marine Engineers, vol. 51, 1943, pp. 115-130.

¹⁶ "Part-Load Characteristics of Marine Gas-Turbine Plants," by W. M. Rohsenow and J. P. Hunsaker, Trans. ASME, vol. 69, 1947, pp. 433-439.

¹⁷ "Development and Testing of a Gas-Turbine Combustor," by A. E. Hershey, Trans. ASME, vol. 69, 1947, pp. 859-867.

¹⁸ Footnote (14), p. 247.

$$\frac{1 - \eta_M}{W_{\text{net}} / (\Delta h \odot \eta_T)}$$

approximates the mechanical loss evaluation in terms of plant specific fuel consumption using the turbine work basis. Its magnitude, in substantial agreement with the above evaluation, is 6.3 per cent.

Aside from convenience in analysis, it appears to the author that mechanical losses in a gas-turbine plant component, other than the compressors or turbines, should be evaluated in terms of the ideal work of the compressor or turbine immediately downstream from the component in question, with the exception of the low-pressure turbine-exhaust piping and regenerator losses in the open-cycle plant which must be evaluated in terms of their effect on the ideal work of the last turbine (5).

Professor Shapiro points out that an efficiency definition should depend on as few parameters as possible and that it should be "functionally significant" rather than thermodynamically significant. The author is of the expressed opinion that the efficiency definition must be both functionally and thermodynamically significant. The author agrees that an expression of over-all efficiency in itself may lead to confusion in appraising the merits of a combustion-chamber design, but is also of the opinion that η_c , η_M , η_0 together are more descriptive than η_c and $\Delta P/P$ alone.

Professor Fax refers to a "breakdown" of the definition of η_{c1} at a magnitude of zero. This breakdown is not apparent to the author as, in accordance with the definition, a magnitude of

$\eta_{c1} = 0$ means, $f_i = 0$. This means, from Equation [7] that $t_p = t_a$. True, this zero of efficiency does not correspond to Professor Fax's "reasonable definition for the zero of combustion efficiency." This latter definition serves to establish, in effect, a thermal enthalpy datum at a temperature t' corresponding to the bulk mean temperature of 1 lb of fuel at t_f and one lb of air at t_a without combustion and assuming no vaporization. This definition has the advantage of removing the ambiguity of the dependence of η_{c2} on the arbitrary selection of the reference state for the heating value term, HV_0 . Consequently, his η_{c3} is preferred to η_{c2} . However, η_{c3} like η_{c2} has many disadvantages when compared to η_{c1} as an expression of the combustion efficiency of a gas-turbine plant combustion chamber.

Professor Fax calls attention to the numerical error in the sample problem evaluating η_{c2} . This should be

$$\eta_{c2} = 0.9174 + 0.0347 = 95.2 \text{ per cent}$$

instead of 95.4 per cent. As he points out, $\eta_{c2} > \eta_{c1}$ except for the limiting case of $\eta_{c1} = \eta_{c2} = 100$ per cent.

Mr. Amorosi agrees in the main with the views presented in this paper. He points out the additional criterion of combustion-chamber performance of uniformity of products outlet temperature, and suggests that a performance factor expressing this effect be formulated.

The author is appreciative of the excellent discussions and criticisms provided by the reviewers.

Temperature Measurements and Combustion Efficiency in Combustors for Gas-Turbine Engines

By WALTER T. OLSON¹ AND EVERETT BERNARDO,^{1,2} CLEVELAND, OHIO

In investigations of the general altitude performance of combustors for aircraft gas-turbine engines at the Cleveland laboratory of the National Advisory Committee for Aeronautics (NACA), the combustors have been instrumented with a large number of thermocouples for determining average temperatures, temperature distributions, and combustion efficiencies. Presented herein is a discussion of the thermocouple installations used, some of the construction details of typical thermocouples, and representative temperature and combustion-efficiency measurements made in combustors. The results of numerous investigations of the general altitude performance of combustors for gas-turbine engines have indicated that air temperatures at the inlet to the combustors can be adequately measured with common bare-wire thermocouples, and that, at the combustor outlet, large variations in temperature distributions, nonuniform velocity distribution, and fluctuations in the temperatures are among the possible sources of errors in measuring the average combustor-outlet temperatures. In addition, the results indicate that the temperatures measured at the combustor outlet, although admittedly not accurate, are generally satisfactory for evaluating the altitude performance of combustors, and that the values of combustion efficiencies obtained from temperature measurements are satisfactory approximations of combustion efficiencies and serve to show the effect of primary variables on combustion efficiency.

INTRODUCTION

EARLY in the investigation of the performance of aircraft gas-turbine engines at the Cleveland laboratory of the NACA, it was found that as engine operating altitude was increased, altitude operational limits were encountered, that is, altitudes were reached above which the combustors could not produce gases of the temperature required by the turbine for engine operation regardless of the fuel flow to the combustor. As a result of these findings and the requirement for high-altitude operation of turbojet engines, a general program was undertaken to investigate the altitude operational limits imposed by combustion and to study methods for improving these limits. This program was necessarily expanded to comprise a more complete investigation of the performance characteristics of combustors for aircraft gas turbines, and the particular and individual effects on combustor performance of operating variables, fuel variables,

and design variables. A number of combustors of several types have been studied to date and this line of endeavor is continuing at present.

For the performance investigations, instrumentation of the combustors included a large number of thermocouples both at inlet and outlet stations not only to measure the temperatures necessary for evaluating the altitude operational limits, but also to determine the temperature distribution at the combustor outlet, to indicate whether or not afterburning occurred and, in addition, to provide the data necessary for calculating combustion efficiency.

It is intended in this paper simply to describe typical thermocouple installations as used in investigating the general altitude performance of combustors at the NACA, to present some of the construction details of these thermocouples, and to illustrate the use of these thermocouples with representative results of combustor temperature measurements and combustion-efficiency measurements. The paper is not intended as a critical analysis of pyrometry with thermocouples; other authors (1 to 5)³ are covering this subject fully. The paper is a presentation of practice found to be satisfactory for examining some performance trends of gas-turbine combustors.

The studies of gas-turbine combustors, in which the thermocouple technique has been used essentially as described, have included work on five annular combustors, four can-type combustors, and two segments of annular combustors. The effect of operating variables on performance has been investigated to a greater or lesser extent on all of these. Many design modifications have been studied on four of the annular combustors and on the two segments. The effect of fuels on combustor performance has been systematically examined in two of the annular combustors and in two of the can-type combustors. This work is mentioned to indicate the extent to which this thermocouple technique has been used on gas-turbine-combustor projects alone.

COMBUSTOR TEST INSTALLATION

A diagrammatic sketch of a section of a typical annular-type combustor test installation is shown in Fig. 1. This installation is selected solely as an example. Combustion air, altitude exhaust, and fuel are supplied to the combustor by the laboratory system. The combustion-air temperature is regulated by electric heaters and a fuel-fired preheater. Combustion-air flow is measured with either a thin-plate or a variable-area orifice, and the fuel flow is measured with calibrated rotameters. Can-type combustors are installed similarly. In any case, care is taken to duplicate exactly the inlet and outlet stations just as they would be in the engine.

THERMOCOUPLE INSTALLATION

Thermocouples are usually installed at four stations (A-A to D-D) as indicated in Fig. 1. The stations are designated as combustor inlet (station A-A, corresponding to compressor outlet of

¹ Flight Propulsion Research Laboratory, National Advisory Committee for Aeronautics, Cleveland, Ohio.

² Jun. ASME.

Contributed by the ASME Fuels Division, Gas Turbine Division, and the American Rocket Society and presented at the Annual Meeting, Atlantic City, N. J., December 1-5, 1947, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 47-A-23.

³ Numbers in parentheses refer to the Bibliography at the end of the paper.

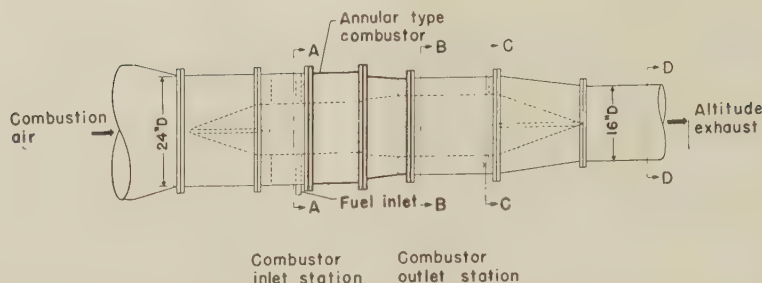


FIG. 1 DIAGRAMMATIC SKETCH OF SECTION OF TYPICAL ANNULAR-TYPE COMBUSTOR TEST INSTALLATION

engine) and combustor outlet (station B-B, corresponding to turbine inlet of engine); thermocouples are also located at stations C-C and D-D to check for afterburning.

The orientation of the thermocouples at each of the stations is illustrated in Fig. 2. All of the thermocouple junctions are lo-

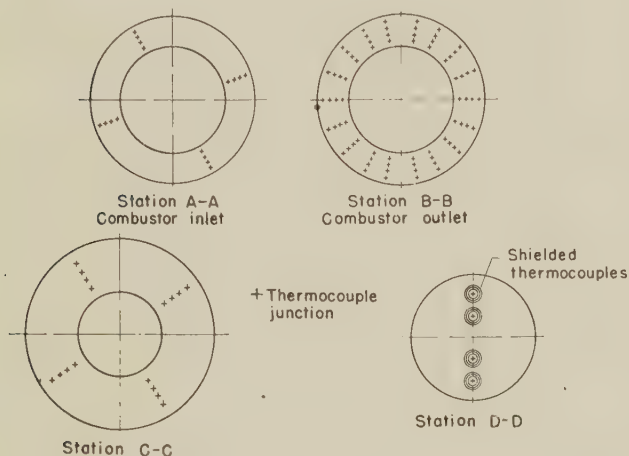


FIG. 2 ORIENTATION OF THERMOCOUPLES IN ANNULAR-TYPE COMBUSTOR

cated at approximate centers of equal areas. The thermocouples are arranged in banks, and the total number of thermocouples used in this installation is noted in Table 1.

TABLE 1 ARRANGEMENT AND NUMBER OF THERMOCOUPLES

	Thermocouple stations			
	A-A	B-B	C-C	D-D
Banks.....	4	18	4	2
Probes per bank.....	4	4	4	2
Total probes.....	16	72	16	4

The construction details of typical thermocouples are illustrated in Fig. 3. All of the thermocouples are generally unshielded, except those at station D-D, which are multiple-shielded in all directions except upstream. The shields of the shielded thermocouples consist of three concentric shells formed from thin sheets of inconel. The thermocouples at the inlet and outlet stations are usually made of 20-gage iron-constantan and chromel-alumel wire, respectively. The wires are threaded through two-hole ceramic insulating rod and sealed within a $1/8$ -in-OD inconel tube. The thermocouple junction is formed by mercury welding. In some instances the junction of the outlet thermocouples has been sealed within conel weld metal to guard against corrosion from catalytic action on the surface of the junction.

In comparison with shielded thermocouples, the unshielded thermocouples are small (blocking a total of about 4 per cent of the cross-sectional area at station B-B), are easy to construct and

to maintain, and are equally as rugged. Unshielded and aspirating-type shielded thermocouples have been compared simultaneously at a combustor outlet under various temperature and velocity conditions; the difference between average temperatures indicated by each type usually was small and inconsistent. For example, during check runs at an average temperature of about 1000 F and an average velocity of 400 fps the difference in average temperatures indicated by the shielded and unshielded thermocouple varied from about ± 5 to ± 10 F.

The thermocouples are connected through multiple switches to self-balancing, indicating potentiometers. The potentiometers for the inlet station (iron-constantan wire) have a range of -100 to $+700$ F, and the potentiometers for the outlet stations have a range from 0 to 2400 F. The accuracy of the potentiometers, as given by the manufacturers, is ± 0.25 per cent of the full scale reading. The instruments are calibrated with the particular thermocouple wire used in each case. In one investigation a specially constructed revolving thermocouple rake in which the thermocouples could be larger and more elaborately constructed was used at the combustor-outlet section, but its use was discontinued at the time because the added complications were not justifiable.

TEMPERATURE MEASUREMENTS

The average temperature at each station is taken as the average of all the thermocouple indications at the particular station. With the thermocouple probes located at centers of equal areas

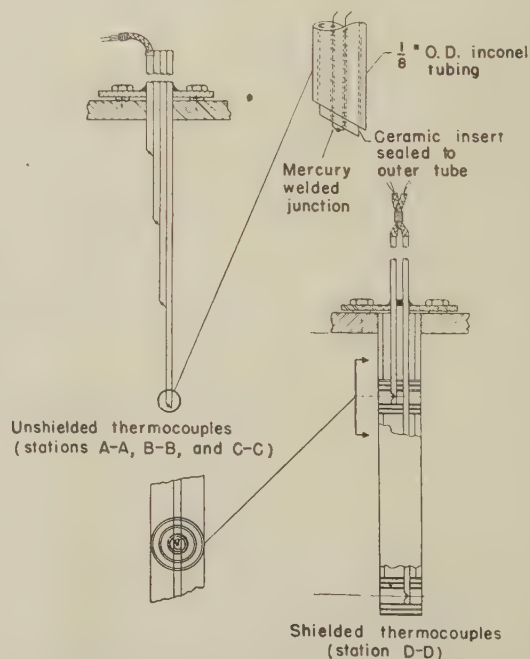


FIG. 3 CONSTRUCTION DETAILS OF TYPICAL THERMOCOUPLES

this provides temperature averages weighted on the mass flow of gases provided of course that the mass flow is uniform at the stations.

In investigating the general altitude performance of combustors, the average combustor-inlet air temperature ranges from approximately -40 to $+300$ F, and the average velocity ranges from about 150 to 300 fps. The temperature and velocity distribution in the pipe conveying air to the combustor may be expected to be fairly uniform; therefore the average temperatures obtained at the inlet station are close to averages weighted on the mass flow of air. The results of an investigation of temperature measurements and of errors involved in measuring temperatures with thermocouples in relatively low-temperature high-velocity air streams are presented by Hottel and Kalitinsky (1). In this reference it is reported that common bare-wire thermocouples with no special stagnation attachments may be considered entirely adequate for temperature measurements in air streams at velocities below about 300 fps.

The local air temperatures actually measured at the combustor inlet are usually within ± 2 F of the average inlet-air temperatures when electric preheaters are used, and within approximately ± 12 F when fuel-fired preheaters are used. The variations in the air velocity over most of the inlet are generally within ± 2 fps during equilibrium flow conditions. There appears to be no particular problem in evaluating adequately the air temperature at the combustor inlet.

At the combustor-outlet stations the average temperatures are usually of the order of 800 to 1400 F, and the average velocities range from about 300 to 600 fps. It is well known of course that temperatures in steady-state high-velocity streams of high-temperature gases cannot be measured with high precision with common unshielded thermocouples, because of the inability of the probes to convert velocity head completely to heat and because of errors arising from such factors as radiation and conduction of heat from the probes (2 to 5).

The problems encountered in measuring temperatures at com-

burntor-outlet stations, however, are not only those encountered in steady-state high-velocity high-temperature gas streams, but include also problems arising from large variations in the temperature distribution, uneven velocity distributions, and fluctuations in the temperatures.

When large temperature gradients are present at the combustor-outlet station, the exact location of a thermocouple probe becomes critical, and a slight change in the probe location may result in a large change in the temperature indicated. This factor, together with any large variation in combustor-outlet velocity distribution, makes it difficult, regardless of the accuracy of the thermocouples, to obtain precise or accurate average temperatures weighted on the mass flow of gases. In addition, the indicated temperatures may fluctuate over a large range if unstable combustion is encountered, thus introducing another source of uncertainty in the observed temperatures.

Typical temperature patterns obtained with thermocouples at the combustor-outlet station for various simulated operating conditions are illustrated in Figs. 4 and 5. The simulated operating conditions are given in terms of engine speed and altitude which are the primary factors in determining the combustor inlet-air conditions. The temperature patterns presented are for annular-type combustors, but in the investigations conducted with different can-type combustors the same general trends have been noted.

Fig. 4 shows the temperature distribution at combustor outlet (station B-B, Fig. 1) for a relatively low-altitude low-engine-speed condition. The temperature distribution under these conditions is fairly uniform; the spread in the local minimum and maximum outlet temperatures from the average outlet temperature (850 F) is about 16 and 24 per cent, respectively, of the average outlet temperature. The largest average radial temperature difference is about 320 F.

The temperature distribution during what is usually termed cycling combustion, that is, where the flames are intermittently

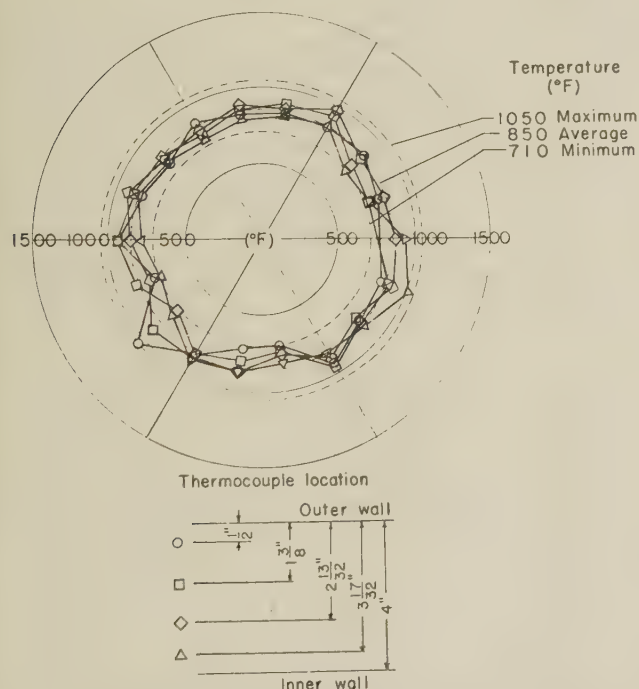


FIG. 4 TEMPERATURE DISTRIBUTION AT COMBUSTOR OUTLET FOR SIMULATED LOW-ALTITUDE LOW-ENGINE-SPEED CONDITION; ANNULAR-TYPE COMBUSTOR

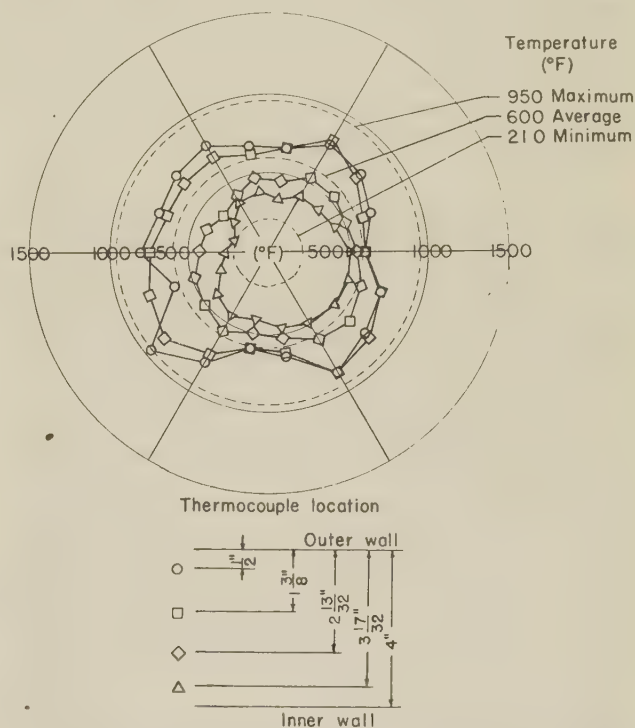


FIG. 5 TEMPERATURE DISTRIBUTION AT COMBUSTOR OUTLET DURING CYCLING COMBUSTION AT SIMULATED LOW-ALTITUDE LOW-ENGINE-SPEED CONDITION; ANNULAR-TYPE COMBUSTOR

extinguished and reignited, for example, in the order of 8 cycles per sec, is shown in Fig. 5 for a condition at the same engine speed, but at an altitude higher than that presented in Fig. 4. The average temperature is of course lower, and the spread in temperatures is appreciably greater than that shown in Fig. 4. The variation in the local maximum and minimum outlet temperature from the average outlet temperature (600 F) is approximately 58 and 65 per cent, respectively, of the average outlet temperature, and the largest average radial temperature difference is about 520 F. During cycling and flickering combustion, the indicated individual temperature at the combustor outlet has been observed to fluctuate 100 F. Although this type of combustion is generally observed only at high-altitude conditions, high-speed motion pictures of combustion show that flickering of lower intensity and higher frequency than that observed at high altitudes prevails in combustors over a large range of altitudes and engine speeds.

Temperature distributions at medium engine speed and high-altitude conditions showed large variations in local temperatures and, at the medium-engine-speed high-altitude conditions, temperature readings indicated that combustion does not occur in the outer section.

The variations in the outlet temperature distribution are usually accompanied by variations in the outlet velocity distribution. Deviations between local maximum and minimum radial velocities at the combustor outlet as large as 70 fps have been obtained when large variations were present in the temperature distributions.

Combustor-outlet temperatures were measured at station B-B, Fig. 1, and compared with the average temperatures measured at stations C-C and D-D, respectively, to check for afterburning. From the data thus derived there appears to be no indication that afterburning occurs. A few of the temperatures at station D-D were somewhat higher than those at station B-B, but these occurred only when large variations were present in the outlet distribution and probably occurred because the four thermocouples at station D-D were not adequate for a representative temperature average. The general agreement of average temperatures at stations C-C and D-D with average temperatures at station B-B is a fairly reassuring check as to the reliability of the averaging method used at station B-B.

COMBUSTION EFFICIENCY

Strictly speaking, combustion efficiency is defined as the ratio of the heat energy liberated during combustion to the heat energy theoretically available from complete combustion. The heat energy actually liberated during combustion is of course difficult to evaluate. At the NACA laboratory, combustion efficiency was approximated (in early work) for simplicity by using the ratio of the actual measured temperature rise across the combustor to the temperature rise theoretically attainable, as evaluated by Turner and Lord (6). The results presented in this paper were calculated on this basis. More recently the preferred method of calculating combustion efficiency is on the basis of the results of a study by Pinkel and Turner (7), although the actual values obtained do not differ appreciably from the temperature-rise ratio. The values of combustion efficiency obtained have usually been satisfactory approximations and have served to show the effect of primary variables on combustion efficiency. Examples of typical combustion-efficiency data obtained in investigating the altitude performance of combustors are presented in Figs. 6 and 7. These results are for annular-type combustors, but the trends shown are similar for can-type combustors.

Lines of constant combustion efficiency, together with a curve of altitude operational limits in a plot of simulated altitude against simulated engine speed are shown in Fig. 6. The com-

bustion-efficiency data below the altitude operational limits were obtained at average combustor-outlet temperatures approximately equal to that required for engine operation at assumed zero flight speed.

The lines of constant combustion efficiency close to the altitude operational limits have the same general shape as the operational limits curve. It has been found that when the altitude operational limits are increased, for example, by design changes in the combustor, the combustion efficiency generally is increased also. At altitudes 5000 to 10,000 ft below the opera-

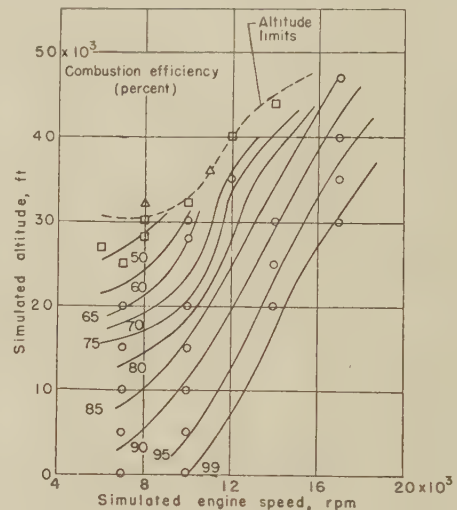


FIG. 6 COMBUSTION EFFICIENCY AS A FUNCTION OF SIMULATED ALTITUDE AND ENGINE SPEED, ZERO FLIGHT SPEED; ANNULAR-TYPE COMBUSTOR

tional-limits curve the combustion efficiency ranges from approximately 50 to 75 per cent; at the low altitudes, the combustion efficiency approaches 100 per cent.

The combustion efficiency decreases with an increase in altitude and with a decrease in engine speed. This effect is further illustrated in Fig. 7, of combustion efficiency plotted against altitude for several engine speeds. Based upon the results of reference (8), in which a detailed study of the effect of combustor-inlet conditions on combustor performance is reported, the decrease of efficiency with increasing altitude and decreasing engine speed is caused by the combustor-inlet-air conditions becoming adverse for combustion with increasing altitude

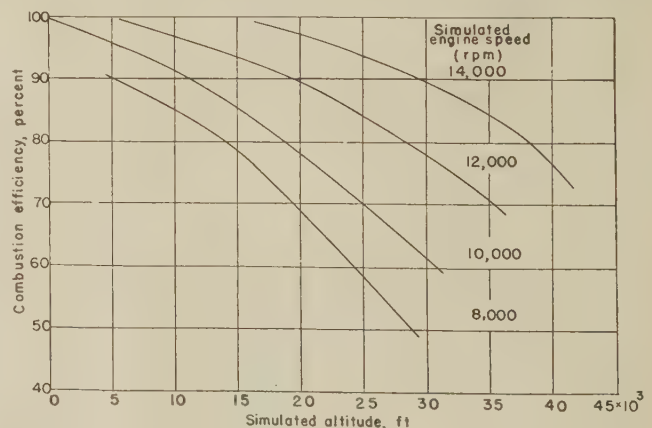


FIG. 7 VARIATION OF COMBUSTION EFFICIENCY WITH SIMULATED ALTITUDE FOR SEVERAL SIMULATED ENGINE SPEEDS (Data from Fig. 6; annular-type combustor.)

and decreasing engine speed. It is also these adverse inlet-air conditions which are responsible for the phenomenon of altitude operational limits.

Finally, it is reassuring to us in using the technique described to note that combustion efficiencies of the order of 98 and 99 per cent are actually measured when combustor operation is of the sort that one anticipates would be nearly 100 per cent efficient. Combustion efficiencies in excess of 100 per cent have not been observed.

CONCLUSION

The results of numerous investigations of the general altitude performance of combustors for gas-turbine engines have indicated the following:

- 1 The air temperatures at the inlet to the combustors can be measured adequately with common bare-wire thermocouples.
- 2 At the combustor outlet, uneven temperature distributions and large radial temperature gradients, nonuniform velocity distributions, and fluctuations in the temperatures are among the sources of errors in measuring average combustor-outlet temperatures.
- 3 The temperatures measured at the combustor-outlet station by means of many bare-wire 20-gage chromel-alumel thermocouples, although admittedly not precise, are generally satisfactory for evaluating the altitude performance of combustors, particularly where the trend of performance with primary variables is of concern.

BIBLIOGRAPHY

- 1 "Temperature Measurements in High-Velocity Air Streams," by H. C. Hottel and A. Kalitinsky, *Journal of Applied Mechanics*, Trans. ASME, vol. 67, 1945, p. A-25.
- 2 "Stagnation Temperature Recording," by W. Wimmer, NACA TM No. 967, Jan., 1941.
- 3 "Measurement of High Temperatures in High-Velocity Gas Streams," by W. J. King, Trans. ASME, vol. 65, 1943, pp. 421-431.
- 4 "A Suggested Method of Measuring High Temperatures in High Velocity Gas Streams at Low Pressure Conditions," by J. E. P. Dunning, RAE TN no. Aero. 1872, S.D. 54, February, 1947.
- 5 Monthly Reports of Progress on the Development of Thermocouple Pyrometers for Gas Turbines, U. S. Department of Commerce, National Bureau of Standards, Washington, D. C.
- 6 "Thermodynamic Charts for the Computation of Combustion and Mixture Temperatures at Constant Pressure," by L. R. Turner and A. M. Lord, NACA TN no. 1086, 1946.
- 7 "Thermodynamic Data for the Computation of the Performance of Exhaust-Gas Turbines," by Benjamin Pinkel and L. R. Turner, NACA Wartime Report originally issued as ARR no. 4B25, Oct., 1945.
- 8 "Effect of Combustor-Inlet Conditions on Performance of an Annular Turbojet Combustor," by J. H. Childs, R. J. McCafferty, and O. W. Surine, NACA TN no. 1357, July, 1947.

Discussion

E. F. Frock.⁴ By using bare thermocouples at the combustor outlet, the NACA has measured temperatures which give approximate values of combustion efficiency adequate to show the general effects of primary operating variables. To accomplish this end it was necessary to use a large number of junctions, located at the centers of annuli of equal area. It is to be noted that the combustor outlet is a most difficult location at which to make reliable temperature measurements, particularly if the combustion reactions are continuing at this station.

Each temperature observed with a thermocouple is merely the

temperature of the measuring junction, and not necessarily the temperature of the gas in which the junction is immersed. In addition to the static temperature of the gas, the following factors are of primary importance in determining the temperature of the measuring junction:

- 1 The rate of heat transfer from the gas to the junction by convection.
- 2 The rates of heat transfer to the junction by direct radiation from the flame, and from the junction by radiation to the colder surrounding walls.
- 3 The gas velocity, and through this the size and shape of the junction.
- 4 The presence or absence of surface reactions catalyzed by the junction itself.

Not all of these effects are in the same direction, and several are interrelated, so that it is impossible to predict their net effect upon the temperature of the measuring junction. However, it may be worth while to consider the possible effects individually, pointing out experimental facts relative to some and items of conjecture relative to others. For this purpose let us assume the following reasonable sea-level operating conditions for a turbojet engine having a compression ratio of 4:1 at rated speed:

Air temperature at combustor inlet, 300 F
 Flame temperature, approximately 3000 F
 True average static temperature at combustor outlet, 1500 F
 Wall temperature at combustor outlet, 1100 F
 Mass flow at combustor outlet, 20 lb/sec ft²
 Combustion efficiency, approximately 100 per cent

Results obtained at the National Bureau of Standards indicate that, under these conditions, a bare junction at the combustor outlet will assume a temperature about 35 F below the static temperature of the gas, provided it receives no direct radiation from the flame and no heat from surface reactions. Therefore such a junction would indicate a combustion efficiency which was low by 35/1200 or about 3 per cent.

It is reasonable to expect that a combustion chamber will operate at nearly 100 per cent efficiency at pressures down to 1 atm abs. If the combustor is operated at the same inlet temperature, but at a pressure of 1 atm, corresponding to an altitude of about 34,000 ft for this engine, the temperatures throughout remain essentially unchanged, but the mass-flow rate is decreased by a factor of 4. Since the rate of heat transfer is nearly proportional to the square root of the mass-flow rate, only one half as much heat reaches the junction per unit time as at sea level. Thus a bare junction, shielded from the flame and in the absence of surface reactions, will read about 70 F low, a figure which has been confirmed by direct experiment. The corresponding error in combustion efficiency is about 6 per cent.

Although not stated directly in the paper under discussion, the impression is conveyed that errors of this magnitude are not considered probable in the NACA results. It must be reasoned therefore that compensating effects, such as radiation from the flame or heat from surface reactions, were present.

As to the energy received by radiation from the flame, we know that this must be proportional to $3460^4 - 1960^4$, which is certainly a sizable number. However, this energy is also proportional to the solid angle over which the junction "sees" the flame, and to the emissivity of the flame, neither of which is known but both of which may be relatively small. At any rate, a bright-yellow flame would surely be a better emitter in the infrared than a pale-blue flame, so that differences in the energy radiated to the junction would be expected from burner to burner and with the operating conditions of a single burner. Since such differences were not reported in the NACA results, it must be inferred either that

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the combined effect of radiation from the flame and surface reaction is too small to measure, that the method is not suited to its detection, or that the results have not been analyzed in the manner necessary to bring out the effect if it was actually present.

It is to be noted that the differences between the static and the total temperatures of the gas under the test conditions are as follows: At 300 fps, 7 F; at 600 fps, 30 F. Assuming a reasonable recovery factor of 0.5 for a bare junction, these errors amount to from 4 to 15 F. The paper does not state whether the observed temperatures were considered as static or as total temperatures. If the former is true, neglecting the velocity effect makes the indicated temperatures too high; if the latter were the case, the indicated temperatures are low by the amounts stated.

Our general experience with multiple bare junctions in a region of continued chemical reaction is that such junctions usually read so high that values of combustion efficiency calculated therefrom are far above 100 per cent, even when junctions farther downstream and shielded from direct radiation from the flame indicate reasonable efficiencies which are in good agreement with those calculated from analyses of the products of combustion.

Without intent to detract from the value of the results obtained by the NACA with the thermometric system described, all of our knowledge and experience indicates that fortune in the form of compensating errors must have played an important role. It should be considered by no means certain that the compensation will be as effective in other installations, and it is the feeling of this reviewer that others should be warned against, rather than encouraged in the use of bare junctions at the outlet of a turbojet combustor, unless the purpose is merely to obtain a very crude idea of the temperature gradients which exist at this location.

STEWART WAY.⁵ Temperature measurement forms the heart and substance of the solution of the problem of evaluating the altitude performance of gas-turbine combustion chambers. Though the method of measurement still leaves something to be desired, a perfectionist attitude would probably have tripled the time of the investigations without altering materially the over-all conclusions.

Some objection might be raised to the use of bare-wire couples to measure temperatures in the 1200-1400 F range. Here, the authors' statement that the use of shielded couples gave not more than a 10 F variation, after averaging, from the unshielded couples, leads to confidence in the probes employed. However, a simple calculation for a bare-wire couple in a 300-fps stream of gas, temperature 1200 F and wall temperature 800 F (0.032-in. wire, 0.050-in. ball couple, 1½-in. leads, 1 atm pressure) gives an error of -54 deg. It is possible that greater accuracy of local temperature measurement might in some instances have led to sufficiently high average outlet temperatures to give over 100 per cent efficiency. In such an event

one would need to introduce more couples into the stream. Ignoring the low-temperature gases in the few tenths of an inch near the walls introduces an error into the results which tends to offset the measured local temperature values being slightly too low.

For the benefit of newcomers in this field, let it be said that the number of temperature stations chosen in some plane in the exhaust-gas stream must be astoundingly large if reliable results are to be obtained. One thermocouple per square inch of cross section is not at all unreasonable. The errors that result from too few probes may be much more serious than those resulting from partial recovery of total temperature, radiation to cool walls, and conduction along the leads.

Variations in local temperature, periodically, with time cause an average local temperature reading somewhat higher than the true mass-weighted mean. This is due to the fact that during the interval in which a cold cloud of gas is passing the couple, the mass flow past the probe is larger than it is in the interval when hot gas is passing the probe. The relationship, if τ is the reciprocal of the absolute instantaneous temperature, is $T_{\text{mean}} = 1/\bar{\tau}$, whereas the measured average temperature, after eliminating other sources of error, is \bar{T} . For example, sinusoidal fluctuations of ± 200 deg superimposed on a time average temperature $\bar{T} = 1660$ deg R correspond to a mass-weighted mean temperature T_{mean} of 1648 deg R. This phenomenon can lead to apparent efficiencies well over 100 per cent if the temperature fluctuations are pronounced, but disregarded in the evaluation.

AUTHORS' CLOSURE

Both Mr. Fiock and Dr. Way are correct in pointing out the factors involved in attaining precise temperature measurements with thermocouples. This was recognized by the authors in citing references 1 to 5. The installation described in the paper may be said to provide a quick and workable solution to a dirty problem. Both average temperature and the temperature profile at approximately the plane of the turbine nozzle are required and are obtained from the system discussed in the paper. Even if shielding had been used and detailed temperature corrections had been made, the fluctuations in flow and temperature at any point in the exit of a combustor preclude a high degree of precision. Temperature variations from thermocouple to thermocouple and, in instances where combustion is faulty and fluctuating, temperature variations on one thermocouple with time are frequently larger than corrections resulting from partial recovery of total temperature, radiation, and conduction. The temperatures measured were considered as total temperatures.

Installations of the sort described have been used by the NACA in a number of different combustors, both annular and can type, and have in all cases resulted in average temperatures that show agreement with average temperatures measured downstream with shielded thermocouples, have indicated combustion efficiency of the order of 98 to 99 per cent when this efficiency is expected, and have never measured above 100 per cent combustion efficiency.

⁵ Manager, Thermodynamics Section, Westinghouse Electric Corporation, East Pittsburgh, Pa. Mem. ASME.

Determination of Gas-Turbine Combustion-Chamber Efficiency by Chemical Means¹

By PETER LLOYD,² PYESTOCK, HAMPSHIRE, ENGLAND

Because of certain limitations of the heat-balance method of determining combustion-chamber efficiency, a more accurate chemical method of analysis has been developed utilizing the gravimetric technique. This method has the added merit of indicating the nature of combustion loss, which at times may be very useful. Various approximate methods, chemical, physicochemical, and physical can be devised, all of which require calibration against an exact and complete chemical determination. Details of the method and its application to the gas-turbine problem are presented.

INTRODUCTION

THE obvious method of determining combustion-chamber efficiency and the one to which all gas-turbine development groups have first turned their attention is the method of heat balance. Heat quantities and temperatures are, after all, the things with which we are mainly concerned, rather than chemical compositions. As a result, the heat-balance method has been developed to a high degree, so that it forms an essential tool in combustion investigations.

It is not the purpose of this paper to discuss the limitations of heat-balance techniques, but since the case for gas-analysis methods turns on these limitations, it is in order to indicate what these are:

(a) *Accuracy.* Heat input is measurable with any degree of accuracy likely to be needed, but heat output, involving the two quantities air mass flow and mean gas temperature in a stratified stream, cannot be so precisely determined. When all possible precautions are taken there is still a probable error of the order of 1½ per cent, and, under the conditions of many development tests, the error is much greater. There are many purposes for which this sort of accuracy is quite good enough, but for others a higher standard must be demanded.

(b) *Nature of the Combustion Loss.* A heat balance indicates the magnitude of the combustion loss, but gives no indication of its nature.

(c) *Applicability to Engine Testing.* The heat-balance method is scarcely applicable to engine tests.

NATURE OF THE PROBLEM

The combustion-chamber outlet has its concentration gradients as well as its temperature gradients; indeed in so far as unburned constituents are concerned these may be even more pronounced. The first requisite therefore is to procure a representative sample. This is a point which was overlooked in Great Britain in some of our earlier work which, as a result, gave us misleading answers.

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² Ministry of Supply, National Gas Turbine Establishment.

Contributed by the Fuels Division and Gas Turbine Power Division and presented at the Annual Meeting, Atlantic City, N. J., December 1-5, 1947, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 47-A-58.

Quenching of the sample to stop further reaction is no difficulty, and any water-cooled sampling device would seem to satisfy this requirement (except that unstable substances such as the peroxides may break down at the sampling-tube temperatures).

Having collected a representative sample and cooled it, there remains the problem of analysis. The combustible constituents which we have so far identified in our combustion gases are carbon monoxide, hydrogen, formic acid, formaldehyde, gaseous hydrocarbons (largely methane), condensable hydrocarbons, carbon.

The presence of acetylene and ethylene was recorded in some early tests but has not been generally confirmed in later and probably more accurate work. Similarly, the presence of peroxides has never been firmly established in normal combustion products.

The relative proportions of these constituents may vary over a wide range. Thus liquid hydrocarbons, which are present in almost negligible proportions under conditions giving high efficiency, can become the predominant component under some circumstances. Similarly, the apparent temperature at which the water-gas equilibrium is frozen can vary considerably.

In the technique of analysis, accuracy must be the primary aim, an objective which is complicated by the high dilution of the gas sample. This is illustrated by the fact that at a representative condition (100:1 air-fuel ratio) the presence of 0.01 per cent by volume of unchanged fuel vapor taken as $C_{12}H_{24}$, represents about 5 per cent combustion loss.

However, accuracy must not be too dearly bought, for if it can only be achieved by excessive complication the usefulness of this result may be much reduced. Five approximate analyses may sometimes be more revealing than one exact one. Speed is therefore a secondary requirement of some importance.

GAS SAMPLING

The fact that steep concentration gradients exist in the exhaust stream of most gas-turbine combustion systems is sufficiently evident. It is illustrated by the wide divergence between the apparent air-fuel ratio, deduced from gas analysis with imperfect sampling methods, and the ratio deduced from measured air and fuel quantities.

The first step toward collecting a representative sample is to withdraw gas from a large number of points across the outlet section. One way of doing this is with a fixed multiple sampling tube drawing individual samples from zones of equal area. Fig. 1 illustrates typical tubes of this kind for use on a circular outlet, the sampling holes being located so as to draw gas from annuli of equal area. A better method is to use a moving tube which continually sweeps the outlet area, and a device of this kind for use on the typical segmental outlet is shown in Fig. 2.

The sampling tube is moved by a cam driven through a flexible coupling by a small electric motor so as to make 15 oscillations per min. With this arrangement the loci of the sampling points are numerous suitably spaced circular arcs. This device insures the provision of sampling points representative of the whole outlet area. However, this is not always a sufficient precaution, for a sample collected in this way is not necessarily a correctly

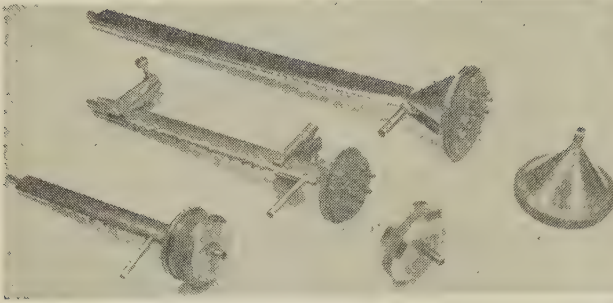
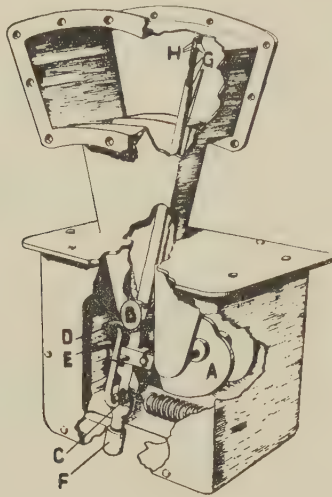
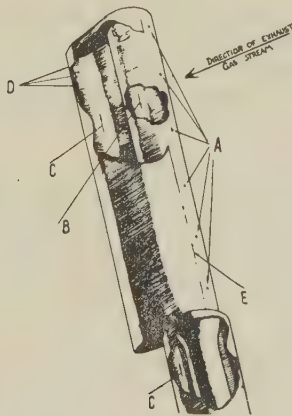


FIG. 1 FIXED SAMPLING TUBES FOR CIRCULAR OUTLETS



- A cam driven electrically via a flexible coupling
- B bearing carrying sampling tube
- C mixing chamber for sample gases
- D sample-gas exit
- E cooling-water entry
- F drain for condensate
- G sampling ports
- H cooling-water exit ports

FIG. 2 AUTOMATIC TRAVERSER FOR SEGMENTAL OUTLET



- A six 0.025-in-diam sampling ports at centers of equal annular segments in exhaust area
- B sample gas
- C water
- D water exit ports
- E weld

FIG. 3 DETAIL OF SAMPLING TUBE FROM AUTOMATIC TRAVERSER

weighted one. A typical sampling technique is to apply a constant suction to the tube, and with most sampling tubes the metering constriction occurs somewhere in the water-cooled part of the system, i.e., the gas is metered cold. Such an arrangement will give a substantially constant sampling rate, and, as the mass flow is not constant across the outlet area, the sample will not be representative.

Consideration of conditions in the outlet stream indicates there is a possibility of getting a truly representative sample, as follows:

Let ρ and v be the density and velocity at a point in the outlet stream.

Let Q_s be the mass flow into the sampling tube at this point. Then the requirement for collection of a representative sample is that

$$Q_s \propto \rho v$$

Now, in the case just considered, the sampling rate is independent of the local velocity and density, i.e.

$$Q_s \propto \rho^0 v^0$$

giving an indeterminate error.

If the dynamic head of the stream is used to cause the flow through the sampling tube, with the metering restriction in the cold part of the system as before, then

$$Q_s \propto \sqrt{1/2 \rho v^2 / g} \times \sqrt{\rho^0} \\ \propto \frac{\rho v}{\sqrt{\rho}}$$

This arrangement approaches more closely to a true sample but is overweighted in terms of the less dense part of the stream, i.e., an excessive proportion of the sample is drawn from the hotter zones. If we retain the dynamic head of the stream as the effective pressure and meter the flow before entry to the cooling passage, then

$$Q_s \propto \sqrt{1/2 \rho v^2 / g} \times \sqrt{\rho} \\ \propto \rho v$$

and the requirement for uniform sampling is satisfied. Metering of the sampling stream before quenching is a condition that is not very easy to meet, but it can be approached fairly closely, and one way of doing this is shown in Fig. 3. On this design the metering orifices are drilled in an uncooled strip lying along the face of the sampling tube. This arrangement has been found in practice to give a reasonably representative sample in spite of most uneven outlet traverses.

An alternative way of achieving the same objective is to place a mixing device in the plane of the outlet so as to create a uniform mixture downstream. From this uniform stream a sample can be drawn using, if necessary, a single fixed tube. This method is naturally permissible only where chemical reaction has already ceased at the outlet plane, but our experience suggests that this condition is well enough satisfied in many cases. A turbine wheel, although the aerodynamicists may not readily admit it, is a fairly good mixer, and sampling from a jet pipe, therefore, may be undertaken with relatively simple devices without involving large sampling errors.

A source of error which is not allowed for in the foregoing argument is that associated with the presence of liquid droplets in the stream, a situation which may arise when heavy fuels are burned. In this condition, a sample representative of the local "streamline" could be obtained only by maintaining the gas velocity in the sampling port equal to the local velocity of the stream. However, such an arrangement would not give a sample representative of the whole gas flow. This difficulty can best be met by the help of an auxiliary mixer. Carbon when present

is assumed to be sufficiently finely divided to follow the gas flow, and the measurement is found to be substantially independent of sampling rate.

A comparison of the air-fuel ratio, deduced from a gas analysis, with that derived from the metered-air and fuel quantities, provides a good if not infallible check on the representative nature of any given sample.

ANALYSIS

A variety of methods have been used or proposed for examining gas-turbine exhausts; in the present paper we need consider only the more important ones.

Cambridge Vacuum Fractionation Method. This method³ was selected in the early days of gas-turbine development as being the most accurate available, especially for dilute mixtures, and a number of samples were examined in this way. This work was invaluable in showing what constituents were present and in providing a standard of comparison.

The method achieves its accuracy by direct measurement (not by difference) of minor constituents and by the avoidance of all liquid reagents. Separation of constituents (or their combustion products) for measurement is achieved by fractional condensation, using a temperature of -80°C to separate water vapor and other readily condensable constituents, and -180°C to condense carbon dioxide. The sequence of operations, greatly simplified, is shown in Fig. 4; another layout of the apparatus is shown in Fig. 5. Neither illustration brings out the specialized nature of the technique,⁴ which involves the use of high vacua, or the complexity of the measuring process which (with the original form of apparatus) required about 2 working days per sample. It was this complexity which led us, in the end, to abandon this method in favor of something simpler.

R. R. Baldwin⁵ has more recently shown that if the hydrogen and formaldehyde in the sample are ignored, then the method can be considerably simplified, reducing it to a level at which a laboratory assistant can complete a single sample in $3\frac{1}{2}$ hr (analyzing for CO_2 , CO , CH_4 , other hydrocarbons, and oxygen). Even so, the technique remains a complex one and the process lengthy. Norrish and Russell⁶ have also introduced a new refinement which may increase the scope of Baldwin's simplified apparatus. This is to deposit on the glass apparatus a monolayer of a hydrophobic substance and so to prevent the adsorption of water vapor. This should make it possible to measure hy-

³ "A Method for the Accurate Analysis of Gaseous Mixtures," by C. H. Bamford and R. R. Baldwin, *Journal of the Chemical Society*, 1942, p. 26.

⁴ "High Vacuum Technique," by J. Yarwood, Chapman and Hall, London, England, 1945.

⁵ Unpublished work with the Shell Company, Thornton Research Center, by R. R. Baldwin.

⁶ "Adsorption of Water Vapour in High Vacuum Apparatus," by R. G. N. Norrish and K. E. Russell, *Nature*, vol. 160, 1947, p. 57.

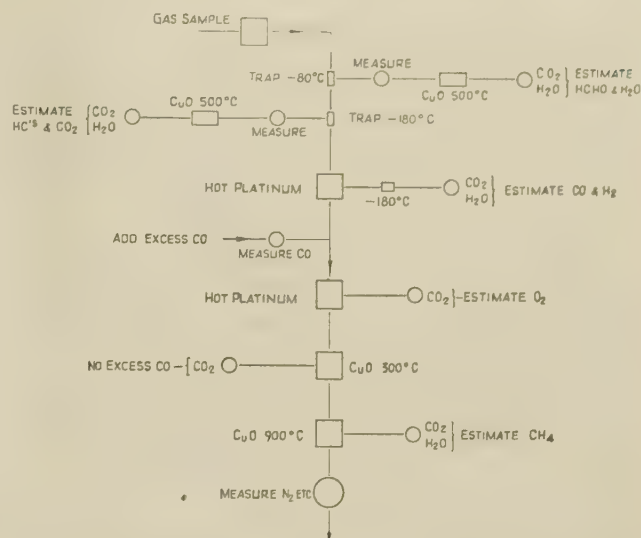


FIG. 4 VACUUM-FRACTIONATION METHOD; SEQUENCE OF OPERATIONS

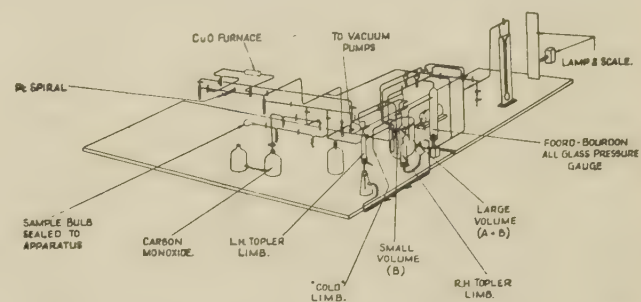


FIG. 5 VACUUM-FRACTIONATION METHOD; LAYOUT OF APPARATUS

drogen concentrations (as water vapor) more accurately and without the sensitive Forord-Bourdon gage used in the original form of the vacuum apparatus.

With the dilute samples characteristic of gas-turbine exhausts, the minor constituents which determine the loss can be estimated to about 0.005 per cent by volume, and Table 1 sets out the errors in the combustion-loss measurement which this degree of accuracy implies. The aggregate possible error due to the constituents measured is shown to be ± 0.73 , but this excludes the possible error due to free carbon which is not measured at all, to unburned fuel which is not effectively measured, and to formic acid which is not differentiated from formaldehyde.

N.G.T.E. Gravimetric Method. In this method the sample is drawn through a trap cooled to -80°C , and the condensate form-

TABLE 1 ERRORS IN CHEMICAL METHODS OF COMBUSTION-LOSS ESTIMATION

Constituent	Cambridge method		N.G.T.E. method	
	Error in estimation by volume, per cent	Error in loss as per cent of heat input at $q = 1/100$	Error in estimation	Error in loss as per cent of heat input at $q = 1/100$
Methane.....	0.005	0.33	0.1 mg CO_2	0.32
Carbon monoxide.....	0.005	0.1	0.1 mg CO_2	0.11
Hydrogen.....	0.005	0.1	0.1 mg H_2O	0.24
Formaldehyde.....	0.005	0.2	0.1 ml of N/100 solution	0.01
Formic acid.....	Not distinguished from formaldehyde		0.1 ml of N/100 solution	0.025
Unburned fuel.....	Not measured effectively		0.1 mg CO_2	0.025
Carbon.....	Not measured		0.1 mg CO_2	0.014
Aggregate possible error	...	0.73 per cent + error from last three constituents	...	0.744 per cent

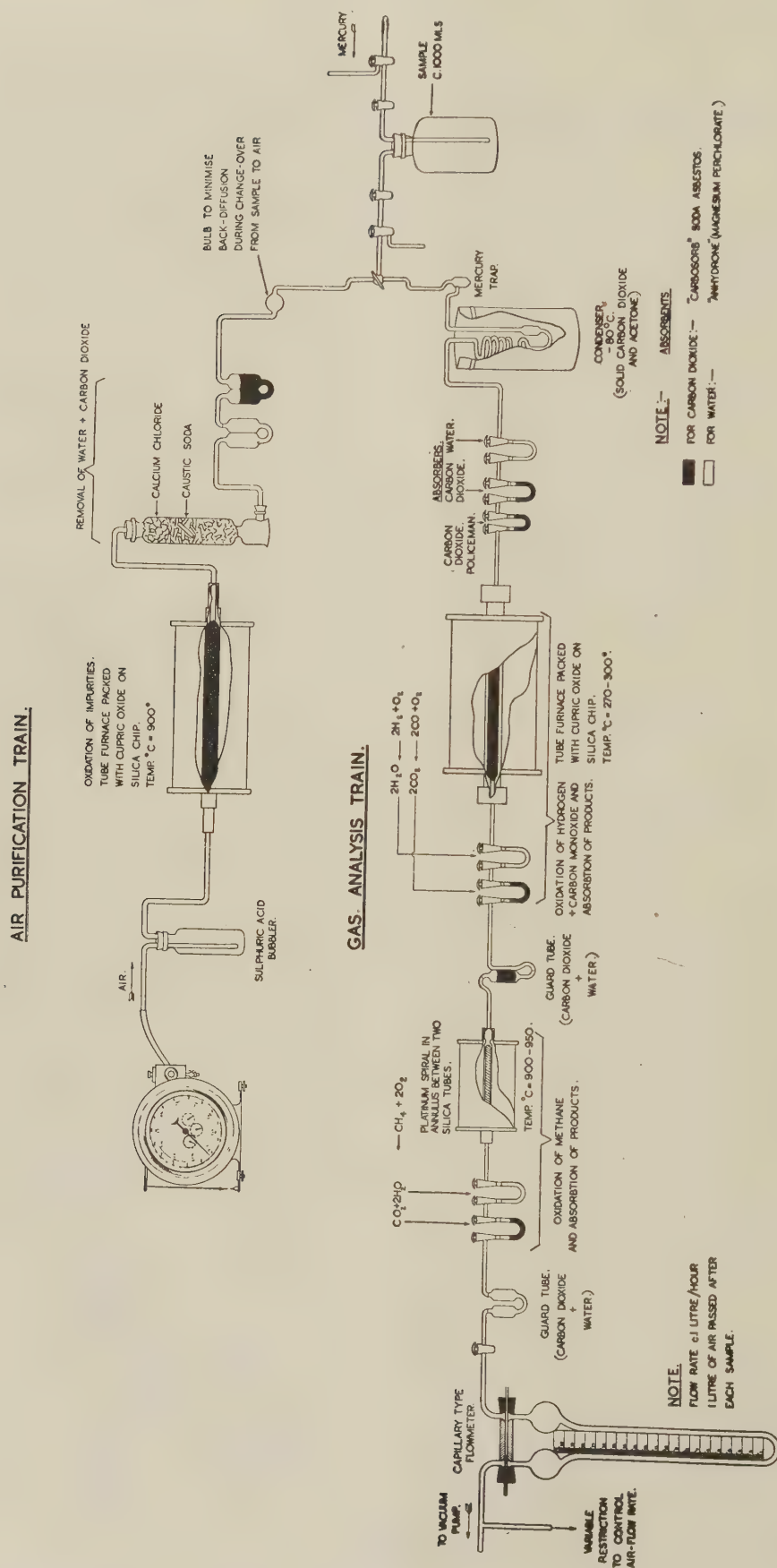


Fig. 6 GRAVIMETRIC METHOD; REACTION TRAIN FOR GASEOUS CONSTITUENTS

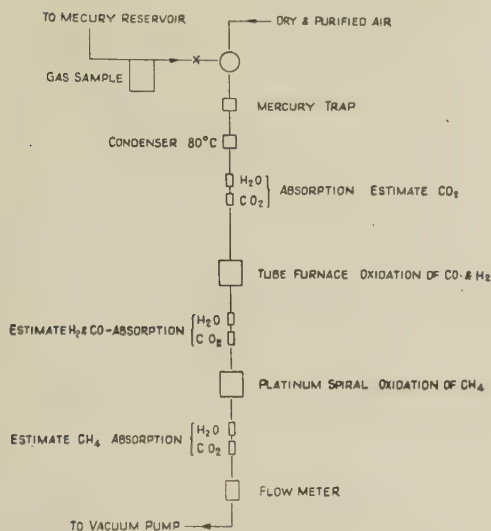


FIG. 7 GRAVIMETRIC METHOD; SEQUENCE OF OPERATIONS

ing there is used for estimation of the total condensable fuel residues; a one-liter sample of the remaining (gaseous) constituents is also collected. These gaseous constituents are passed through a reaction train, Figs. 6 and 7, in which they are separated by selective oxidation, followed by absorption of reaction products. This disposes of carbon dioxide, carbon monoxide, hydrogen, and gaseous hydrocarbons (methane is the only one found to be present). The condensable fuel residues from the freezing trap are burned in a stream of oxygen and estimated as carbon dioxide. This result needs to be corrected for other condensable constituents if present. Free carbon is collected in a miniature Gooch crucible, mounted in a water-cooled sampling head; it is estimated in the form of carbon dioxide after combustion.

The other condensable constituents, formaldehyde and formic acid, are measured by titration after separate absorption processes.

Chemical aspects of these measurements are fully described in a recent paper⁷ by J. J. Macfarlane, who has been responsible for the development of the method.

The accuracy of the method depends mainly on the quantity of the sample drawn and the accuracy of the balance used. At present we have a balance sensitive to 0.1 mg, and we take 1-liter samples for the main reaction train, 5-liter samples for formaldehyde and formic acid, 10-liter samples for unburned fuel, and 12-liter samples for carbon. Assuming further that the titrations have an accuracy of ± 0.1 ml of N/100 reagent, then the possible errors in efficiency estimation at air-fuel ratio 100:1 are those listed in Table 1. The errors listed in this table are, for either method, proportional to the air-fuel ratio.

Assuming the simultaneous analysis of a batch of samples with two reaction trains in operation, the time required for each complete analysis is about 2 man-hours. Using a single reaction train or analyzing individual samples takes correspondingly longer.

Approximate Chemical Methods. For quick approximate analyses, various methods are possible. An obvious one is to estimate efficiency in terms of the ratio.

$$\frac{\text{Net carbon}}{\text{Gross carbon}} = \frac{\text{Mols of carbon present as CO}_2}{\text{Mols of total carbon present}}$$

The sample for indication of net carbon is prepared by drawing

through a freezing trap at -80°C to remove formaldehyde and formic acid. That for total carbon is passed through a furnace containing copper oxide at 950°C , in which all combustibles will be burned. It is easy to arrange for simultaneous sampling through the two circuits in parallel. Alternatively, it may sometimes be preferable to determine the ratio

$$\frac{\text{Mols of carbon unburned}}{\text{Mols of total carbon}}$$

as may be done, with a certain error, by drawing the numerator sample through soda lime to remove existing CO_2 , through a furnace to oxidize the unburned constituents, and then to the analyzing apparatus for estimation of the mols of carbon unburned. The error involved is that formic acid and formaldehyde will be absorbed with the initially present CO_2 , while some unburned fuel may also be lost.

Analysis may be effected by the well-known Haldane apparatus or by various other means. For speed, a physical method may be preferred, e.g., the measurement of the hydrogen-ion concentration in a sodium-bicarbonate solution saturated with the gas under examination. This method, which was first proposed by Wilson, Orcutt, and Peterson⁸ has been developed further by Baldwin⁴ for this purpose. A standard glass electrode is used, fitted in the solution through which the gas sample is bubbled, the second electrode being a reference calomel electrode in saturated potassium chloride. This device gives a direct indication of pH from which CO_2 percentage is simply calculated. The accuracy is ± 2 per cent of the CO_2 reading over the whole range, and the time required to reach equilibrium is a few minutes.

To equate the ratio net CO_2 -gross CO_2 to combustion efficiency would be to assume that the heat of combustion of the unburned constituents is the same as that of the fuel needed to produce the same quantity of CO_2 , which obviously is not justified. Experience shows that the ratio of these heats of combustion, and hence the ratio of true combustion loss to per cent carbon unburned, varies substantially with a mean value in the region of 0.65. For the purpose of the approximate method we therefore can use the generalization

$$\text{Combustion loss per cent} = 0.65 \times \frac{\text{Carbon unburned}}{\text{Total carbon}}$$

The accuracy of this method will be discussed later. The time required for sampling and analysis, using the Haldane apparatus, is about $1/2$ hr; using the pH meter, about 10 min.

Approximate Physical Methods. Various other approximate methods are possible and may have a certain application; none of these is as fully developed as the chemical methods, but they deserve mention because of their potential speed and simplicity.

There are, for example, the methods depending on the combustion of some or all of the unburned constituents on a catalyst, and the measurement of a heat quantity. Ideally, the catalyst should oxidize all the unburned gases and vapors and so give a direct measure of the heat of combustion of the exhaust gas, and hence of the combustion loss. This ideal has not, to the author's knowledge, been achieved, the chief problems being the ignition of the methane and the avoidance of poisoning on the catalyst surface. There is no reason, however, to consider these problems insoluble.

A second best is a catalyst which will deal reliably with a representative part of the unburned gas and vapor. It has been

⁷ "The Application of Chemical Analysis to the Investigation of Gas Turbine Combustion Problems," by J. J. Macfarlane, Proceedings of the International Congress of Pure and Applied Chemistry, 1947.

⁸ "Determination of Carbon Dioxide in Gas Mixtures," by P. W. Wilson, F. S. Orcutt, and W. H. Peterson, *Industrial and Engineering Chemistry, Analytical Edition*, vol. 4, 1932, p. 357.

found in Great Britain that if the gas sample is suitably purified to remove constituents which poison it, a hot platinum wire will provide a suitable surface and will give an indication of the combined concentration of carbon monoxide and hydrogen. Methane is not oxidized, and the purifying process removes the heavy hydrocarbons, formaldehyde, and formic acid. The mechanism of the instrument⁹ is that the heat released at the surface of the wire raises its temperature, so changing its resistance and throwing a bridge out of balance. There are two stages of approximation here, namely, (1) the instrument does not respond equally to carbon monoxide and hydrogen so that with a variable CO-H₂ ratio, as we may have, the calibration is arbitrary; (2) the heat loss in (CO + H₂) is not necessarily a fixed proportion of the whole. This point will be further examined in a later paragraph in which some results are presented.

Another physical method which may prove to be important is that based on infrared absorption; the principle is as follows: Two matched low-temperature elements send their heat radiation through two absorption chambers into receivers, both of which are filled with the pure gas to be estimated. One absorption chamber is filled with the gas sample; the other is a comparison chamber which is usually filled with air. Both radiation paths are periodically interrupted by a shutter. Radiation passing through the gas sample is partly absorbed, and this effect reduces the heat reaching the receiver; both absorptions are associated with the characteristic infrared spectrum of the gas being estimated. As a result, changes in the heat picked up in the receiver constitute a specific measure of the concentration of the gas which is being estimated.

The purpose of the second channel is to balance out the slow and unselective heating of the measuring-chamber walls and so to make the instrument respond only to changes in gas temperature. The two measuring chambers are separated by a fine membrane which forms one plate of a variable condenser. Changes in gas pressure in the two halves of the receiver thus cause changes in capacity of the little condenser; these are converted into potential changes, rectified, amplified, and indicated on a millivoltmeter.

This system can be made extremely sensitive and can indicate to 2 per cent or less of a full-scale deflection corresponding to 0.2 per cent by volume of the gas being measured. Some care may be needed to maintain consistent accuracy. The indication is virtually instantaneous, and, for a single gas such as carbon monoxide, it should be very good. However, measurement of a single constituent is hardly enough for gas-turbine applications, so that, for this system to become effective, a more complex arrangement will be necessary. Probably the best way of using it would be in conjunction with an approximate chemical method, e.g., for indication of CO₂ per cent formed by combustion of the unburned constituents in the gas.

RESULTS

Table 2 shows a set of results obtained¹⁰ by the gravimetric method on an experimental combustion chamber of the aero-engine type, i.e., burning distillate fuels at a high combustion intensity. The test results cover a fair range of conditions and illustrate the changes in the proportions of the various loss constituents which may take place. Thus the carbon monoxide/hydrogen ratio varies widely and, in an apparently indiscriminate fashion, unburned fuel may be negligible but may equally well be the predominant constituent of the combustion loss. Formaldehyde and formic acid are of secondary importance in all these

⁹ Manufactured by the Cambridge Instrument Company, 13 Grosvenor Place, London, S.W. 7, England.

¹⁰ By J. J. Macfarlane, F. H. Holderness, and H. Smith at the National Gas Turbine Establishment, England.

TABLE 2 EXAMPLES OF COMBUSTION LOSSES DEDUCED FROM ANALYSES

Experiment	Metered A/F	A/F by gas analysis	Air velocity, fps	Air temp, deg C	Combustion loss $\sigma/0$					Errors in approximate determinations of com- bustion loss; expressed as per cent of heat input 0.65 X		
					H ₂	CO	CH ₄	C	Fuel	HCHO	HCOOH	Total
Comparison of fuels (T scheme Mark IV with 2 rows primary holes)	Reference kerosene	64.3	242	52	0.3	1.5	<0.1	Nil	0.1	<0.1	<0.1	1.9
		76.2	248	55	0.2	1.2	<0.3	Nil	0.1	<0.1	<0.1	1.8
		112.4	253	57	0.9	3.5	<0.1	Nil	<0.1	<0.1	<0.1	4.5
		164.0	252	58	1.6	5.5	<0.1	Nil	0.2	0.1	0.2	7.6
		63.2	242	63	<0.1	1.5	<0.1	<0.1	0.2	<0.1	<0.1	1.8
	Reference gas oil	85.4	254	63	0.9	1.4	<0.1	0.1	0.2	0.1	0.1	2.8
		116.8	257	62	1.1	3.5	<0.1	<0.1	<0.1	0.1	0.1	4.8
		137.6	259	62	1.3	4.3	<0.1	0.2	0.3	0.2	0.2	6.5
		62	246	54	0.5	2.7	<0.1	0.1	0.1	0.1	0.1	3.5
		90.7	254	57	0.7	3.9	<0.1	<0.1	1.6	0.2	0.1	6.5
Effect of pressure (T scheme Mark IV with one row of primary slots)	4 atm	142	259	59	0.7	5.9	<0.1	<0.1	2.0	0.2	0.2	9.0
		61.6	250	99	0.3	0.8	0.2	1.3
		84.7	250	106	0.4	0.5	<0.1	0.9
		113.2	250	108	0.5	1.2	<0.1	1.7
		133.3	250	108	0.9	1.3	<0.1	2.2
	2.2 atm	53.5	250	104	0.2	1.0	0.2	...	<0.1	0.1	<0.1	1.4
		87.0	250	106	0.2	1.3	<0.1	...	<0.1	1.5
		112.8	250	106	0.5	2.5	<0.1	...	<0.1	0.1	...	3.4
		134.5	250	82	0.9	3.0	<0.1	...	0.3	0.1	0.1	4.2
		58.8	252	90	0.7	3.0	<0.1	...	<0.1	<0.1	<0.1	2.1
1 atm	1 atm	51.9	252	94	0.1	1.9	<0.1	Nil	0.1	<0.1	<0.1	2.1
		82.0	259	98	0.8	3.7	<0.1	Nil	0.3	0.1	0.1	4.9
		87.2	267	103	1.0	5.6	<0.1	Nil	0.8	0.2	0.1	7.7
		125	267	103	1.0	5.6	<0.1	Nil	0.8	0.2	0.1	7.7
		51	158	61	0.9	4.8	0.5	...	9.4	0.4	0.2	16.2
	1/2 atm	61.7	158	62	1.7	5.9	0.2	...	12.8	0.5	0.1	21.2
		80.7	158	60	1.0	6.0	1.4	...	18.9	0.3	0.3	27.9
		62.5	158	62	1.7	5.9	0.2	...	12.8	0.5	0.1	21.2
		80.7	158	60	1.0	6.0	1.4	...	18.9	0.3	0.3	27.9
		62.5	158	62	1.7	5.9	0.2	...	12.8	0.5	0.1	21.2
1/2 atm	1/2 atm	62.5	158	62	1.7	5.9	0.2	...	12.8	0.5	0.1	21.2
		80.7	158	60	1.0	6.0	1.4	...	18.9	0.3	0.3	27.9
		62.5	158	62	1.7	5.9	0.2	...	12.8	0.5	0.1	21.2
		80.7	158	60	1.0	6.0	1.4	...	18.9	0.3	0.3	27.9
		62.5	158	62	1.7	5.9	0.2	...	12.8	0.5	0.1	21.2
	1/2 atm	62.5	158	62	1.7	5.9	0.2	...	12.8	0.5	0.1	21.2
		80.7	158	60	1.0	6.0	1.4	...	18.9	0.3	0.3	27.9
		62.5	158	62	1.7	5.9	0.2	...	12.8	0.5	0.1	21.2
		80.7	158	60	1.0	6.0	1.4	...	18.9	0.3	0.3	27.9
		62.5	158	62	1.7	5.9	0.2	...	12.8	0.5	0.1	21.2

analyses, but this is not always the case. Carbon formation increases with heavier fuel (gas oil) and carbon-rich fuel (tetralin); it also increases with combustion-chamber pressure, although this point is not brought out by Table 2.

The sampling error is appreciable in many of these tests, largely because the combustion chamber on which this work was done was not a fully developed type and had a very bad outlet traverse. The improved sampling tube was used only on the first group of samples (tests on alternative fuels).

The last four columns of the table give some indication as to how approximate methods might work out in practice. Measurement of carbon monoxide alone is seen to be of very little value. Even in the region of high efficiency it indicates a loss which differs substantially from the true value, and in the region of low efficiency this error rises to over 20 per cent. Measurement of carbon monoxide plus hydrogen is rather better in the high-efficiency region, but still shows a big error in the low-efficiency condition.

The usefulness of these methods is evidently limited to regions of high efficiency and even here they will give indications rather than measurements of combustion loss.

The carbon-balance method is rather more consistent but even this indicates a widely varying proportion of the true loss. A refinement of the carbon-balance method would be to use it in conjunction with an estimation of carbon monoxide, e.g., by selective oxidation in a low-temperature copper-oxide furnace. This would call for measurement of three CO_2 concentrations, preferably by one of the following rapid physical methods:

- (a) Total CO_2 after oxidation of all unburned constituents.
- (b) CO_2 formed by oxidation of CO .
- (c) CO_2 formed by oxidation of all unburned constituents.

Total combustion loss would then be calculated as follows:

$$\text{Total loss} = \left(\frac{\text{CO}_2 \text{ from CO}}{\text{Total CO}_2} \times \frac{\text{Heat of combustion of CO}}{\text{Heat of combustion of fuel}} \right) + \left(\frac{\text{CO}_2 \text{ from total unburned const} - \text{CO}_2 \text{ from CO}}{\text{Total CO}_2} \right)$$

The loss which would be indicated in this way is given in the last column.

It will be realized that while these test results cover a fair range of conditions, any generalized conclusions which are drawn

from them must be qualified. Other designs and other fuels may give substantially different results.

CONCLUSIONS

The most accurate method available for determination of combustion efficiency is the chemical one using the gravimetric technique. The method has the incidental merit of indicating the nature of the combustion loss, information which may be very helpful in diagnosis. Various approximate methods, chemical, physicochemical, and physical can be devised, and there are applications for several of them depending upon circumstances. All the approximate methods require calibration against an exact and complete chemical determination.

Whatever analytical method is used, the collection of a representative sample is a first essential, and analyses which reveal a discrepancy between the calculated air-fuel ratio and that deduced from air and fuel measurement must be treated with reserve.

ACKNOWLEDGMENTS

The author's thanks are due to various colleagues, especially to J. J. Macfarlane whose work has been freely quoted, and also to the Ministry of Supply, London, England, for permission to publish this paper.

Discussion

W. J. KING.¹¹ This paper represents an excellent contribution to the subject of gas-turbine combustion. It seems to the writer that an added merit of the chemical method proposed by the author lies in the fact that it provides a means for obtaining an independent check upon combustion-chamber efficiency as determined by conventional methods. All of us who have spent any considerable amount of time working with heat have been impressed by the liability of error in the measurement of the flow or generation of heat, when only a single method is employed. The error may sometimes be consistent and therefore all the more difficult to detect. Accordingly, it is always helpful to have an alternative method for measuring quantities of heat, as by a heat balance, and the author's method offers a means for obtaining such a check.

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Mechanisms of Combustion and Their Relation to Oil-Burner Design

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The combustion of fuel oils is a very complex process from the viewpoint of the kineticist. In any attempt to treat this subject theoretically the over-all process must be broken down into a group of consecutively and frequently concurrently occurring chemical and physical changes and each of these analyzed in turn. From an integration of the conclusions obtained by this means a more comprehensive conception of the factors involved in the burning of hydrocarbon fuel oils may be attained. In this paper an attempt will be made to show what processes are involved, and means by which this information may be utilized in the design of burners will be indicated.

THEORETICAL CONSIDERATIONS

INASMUCH as the chemical reaction wherein the oil unites with oxygen to liberate heat and combustion products is of primary interest, certain aspects of the kinetics of chemical reactions will be discussed briefly (1).² The combustion of fuels generally occurs in the vapor phase; consequently, we are concerned with the kinetics of gaseous reactions. The molecules of gas (whether hydrocarbon, oxygen, or inert) are in a state of constant motion, the velocities being distributed according to the Boltzmann distribution law. Each gaseous molecule is continuously undergoing collisions with other molecules, but, in general, these collisions are elastic and the colliding molecules separate unchanged. If the sum of the energies of two or more colliding molecules exceeds a certain minimum value, however, and if the excess energy is properly distributed in those molecules the collision becomes inelastic and chemical reaction occurs between the colliding molecules. The reaction rate is dependent on temperature in two ways; first, in relation to the frequency of collision, which is proportional to the square root of the absolute temperature; and second, to the percentage of the molecules possessing energy in excess of the minimum value required for reaction to occur (the "activation" energy), which is an exponential function given by the expression $e^{-E/RT}$, wherein T is the absolute temperature and E and R are constants.

Chemical reactions occur by one of two mechanisms, namely, by a thermal or by a chain mechanism. In thermal reactions molecular species only are involved. The energy of activation is distributed among translational, rotational, and vibrational degrees of freedom. Electronic forces such as residual valency in general are not involved. Energies of activation and rates of reaction are in general rather easily measured. Reactant molecules dissipate the excess energy liberated in the reaction non-specifically; frequently the dissipation of the energy of reaction raises the temperature of the reacting mass, resulting in an acceleration of the reaction. If the initial temperature exceeds a

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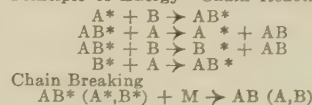
² Numbers in parentheses refer to the Bibliography at the end of the paper.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 47-A-34.

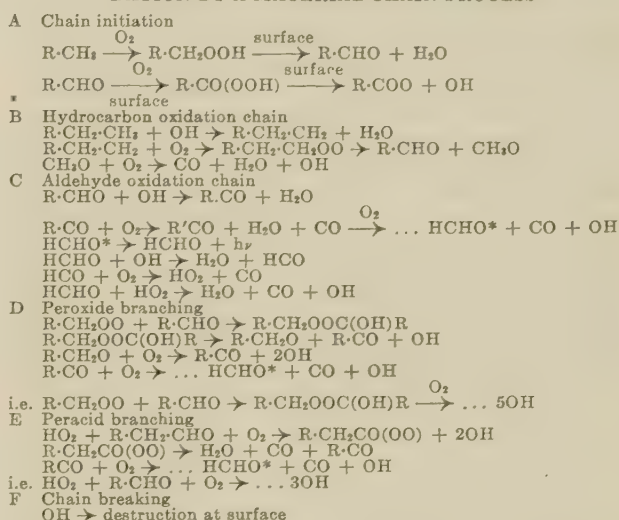
TABLE 1

Principle of Energy—Chain Reactions



NOTE: A and B are the reactants, AB the resultant molecules. Starred molecules represent energy-rich molecules. M represents an inert molecule (or the wall).

TABLE 2 POSTULATED MECHANISM OF HYDROCARBON OXIDATION BY A MATERIAL CHAIN PROCESS¹



NOTE: HCHO* represents the activated formaldehyde molecule to which the "cool flame" radiation is ascribed.

certain value the reaction will accelerate until the rate becomes infinitely large; a thermal explosion is said to occur in this case. Preceding the explosion, however, a rise in temperature and an acceleration of the reaction, frequently measurable, inevitably occur.

In the second type of reaction mechanism, known as the chain mechanism, the initial reaction is invariably followed by a chain of successive reactions. In some reactions the resultant molecules transfer their excess energy more or less specifically to one or another of the reactant molecules and thus give rise to a series of reactions initiated by a single molecule having energy in excess of the activation energy. The principle involved is illustrated in Table 1. In such a reaction the chain carriers are energy-rich molecules. Consequently it is known as an energy-chain reaction. Other chain reactions include as reactants atoms and metastable molecular species (radicals). This type of reaction is illustrated in Table 2. In such reactions the over-all rate is proportional to the concentrations of chain carriers (atoms or radicals) and consequently is affected by the rate of formation and destruction of the carriers. Frequently one step in the re-

² Mechanism proposed by B. Lewis and G. von Elbe, "Combustion, Flames, and Explosions of Gases," Cambridge University Press, London, England; The Macmillan Company, New York, N. Y., 1938, chapter 4.

action gives rise to more than one chain carrier and the chain is said to branch. If the rate of chain initiation and branching exceeds the rate of destruction of chain carriers the reaction rate increases rapidly and approaches infinity, resulting in a "branched chain" explosion. Chain reactions, whether of the energy or material type, cannot readily be analyzed by the method of Arrhenius with the end of obtaining activation energies, inasmuch as the reaction invariably involves a number of steps.

THE OXIDATION OF HYDROCARBONS

The oxidation of hydrocarbons, particularly of the gaseous and more volatile liquid hydrocarbons, has been studied extensively and has been shown to proceed by a chain mechanism. Most of the studies have been made at lower temperatures (between 400 and 1100 F) and at atmospheric or subatmospheric pressures where the reactions proceed slowly. Chain initiation occurs at the surface by a heterogeneous reaction at low temperatures, but homogeneous gas-phase reactions presumably initiate chains at higher temperatures, especially in the oxidation of higher paraffins and alkyl-substituted benzenes. The initial reaction product is generally believed to be a peroxide. Further reaction of the peroxide molecules produces hydroxyl (OH) radicals. Hydroxyl, alkyl, and RCO radicals (the latter two produced by interaction of the OH radicals and oxygen molecules with hydrocarbon molecules) constitute the principal chain carriers of the oxidation process. One of the proposed oxidation mechanisms involving these radicals is presented in Table 2. Chain branching presumably is an important part of hydrocarbon oxidation; two different mechanisms of branching have been postulated, as shown in Table 2, one relating to low temperatures, the other to a high-temperature range (2). That two different mechanisms exist seems to be indicated from the temperature-pressure self-ignition characteristics of hydrocarbons, an example of which is shown in Fig. 1.

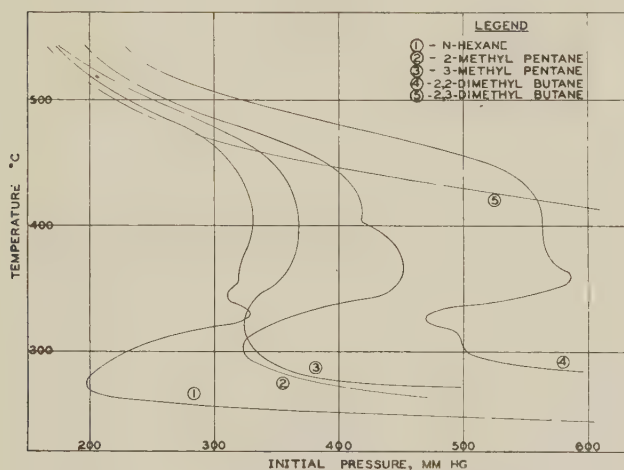


FIG. 1 IGNITION CURVES OF STOICHIOMETRIC MIXTURES OF HEXANE AND OXYGEN. (HEIPLE, UNPUBLISHED DATA)

In the slow oxidation of hydrocarbons the concentrations of the chain carriers, whether OH, R, or RCO, are vanishingly small, and even in explosive reactions in experimental vessels of conventional size the exhaustion of reactants prevents the attainment of a detectable concentration of radicals and atoms. In the flame produced by the burning of hydrocarbons (such as occurs with a Bunsen burner, a kerosene lamp, or an oil burner), however, the concentration of radicals and atoms becomes appreciable in a relatively narrow zone, the flame front. The radicals formed in the flame front may or may not be the same as those in the slow oxidation reaction. There is evidence that

different mechanisms are involved in slow oxidation and in the flame. This may be expected, inasmuch as the temperatures involved are of different orders of magnitude. The rate of liberation of energy in the flame is very high (as is evident from the fact that visible and ultraviolet radiation is emitted), and reactions may occur for which the necessary activation energy is not available at lower temperatures. We would, accordingly, expect differences in the mechanism of oxidation in the slow oxidation of hydrocarbons and in flames.

The rapidity and the completeness of the reactions involved in the burning of hydrocarbons and other fuels with the appearance of flame have precluded to the present any detailed elucidation of the reaction mechanism. Our present knowledge has been obtained largely from spectroscopic studies of flames (3). Broadly speaking, three types of flames may be recognized, namely, "blue" flames, "green" flames, and "luminous" flames. The first two types of flames are generally obtained when lean through stoichiometric to slightly rich mixtures of saturated hydrocarbons and air or oxygen are burned, the third in the burning of rich mixtures of aromatic or highly unsaturated hydrocarbons (e.g. acetylene) and oxygen or air. The spectroscope reveals that the blue flame emits lines and bands ascribed to OH and CH, as well as the so-called "Ethylene Flame Bands," sometimes ascribed to HCO. These lines and bands are in the blue-violet and ultraviolet portion of the spectrum. The color of the green flame arises from the "Swann" bands which occur in the green portion of the spectrum and are ascribed to the radical C_2 . Luminous flames, which show a continuum over most of the visible spectrum, owe their luminosity to incandescent carbon particles. Dependent on the temperature of the flame, the light emitted from the luminous flame may vary from red-orange through yellow to white. The color of the emitted light, however, has no apparent relation to the mechanism of the reaction.

In relation to flame color it should be noted that any hydrocarbon can be made to burn with any one of the three types of flames, the character of the flame being more dependent on the fuel-air ratio and the homogeneity of the mixture than on the molecular construction of the hydrocarbon. Thus acetylene or benzene can be made to burn, under a rather narrowly limited range of conditions, with a blue flame, and methane can be made to burn with a luminous flame.

The occurrence of such radicals as CH and C_2 in the flame indicate that other modes of reaction occur in flames than in the slow oxidation. There is some reason to believe that hydrogen atoms may be "stripped" from a hydrocarbon molecule, with attendant formation of C_2 radicals. This is indicated by the appearance of the Swann bands in flames of mixtures slightly richer than stoichiometric. Deposition of carbon has been ascribed to polymerization of C_2 radicals (4). The highly luminous flames obtained with rich mixtures with their tendency to smoke and form soot strengthens this assumption. It seems probable that oxygen atoms play a significant part in the reactions in the flame front, inasmuch as the energy levels are sufficiently high to cause dissociation of the oxygen molecule. The presence of CH and C_2 spectral bands would suggest a tendency to split off segments of the molecule by breakage of the C—C bond at certain stages in the oxidation process, another reaction that is not considered plausible at lower temperatures. These and similar considerations must be evaluated in attempting to formulate a mechanism for oxidation in the flame, a project which would be far beyond the scope of this paper.

COMPETING REACTIONS INVOLVING HYDROCARBONS

Although the combustion mechanism is of such complexity that the processes involved have not yet been determined and

set in order, an additional complication arises through the occurrence of a number of reactions involving only the hydrocarbon molecules. The reactions involved are those commonly designated as cracking, polymerization, and isomerization. These three types of reaction are theoretically separable, but in practice they are generally insolubly interlinked. Cracking implies the scission of C—C or C—H linkages, with the resultant formation of hydrocarbon molecules of lower molecular weight and frequently of different molecular type. Also combined and frequently classified with cracking, is the dehydrogenation of hydrocarbons, such as occurs when a hydrocarbon molecule loses two or more atoms of hydrogen with formation of double or triple bonds. Polymerization reactions involve the fusion of two or more molecules to form a higher molecular-weight molecule, and isomerization is the process wherein the hydrocarbon molecule undergoes a change in molecular structure without an attendant change in molecular weight. Under certain conditions these three types of reactions occur concurrently with the oxidation processes, and to these reactions certain of the problems of the burning of fuel oils are ascribed. Hydrocarbons of different types show markedly different tendencies to crack; accordingly in the discussion to follow the several hydrocarbon types will be considered separately (5).

Paraffin hydrocarbons, such as open-chain hydrocarbons containing only single C—C bonds, react principally by scission of C—C bonds to form paraffins and olefins of lower molecular weight. It appears that the hydrocarbon chain may break at any point, since analysis of the products from pyrolysis of the lower paraffins generally reveals the occurrence of at least one hydrocarbon having up to $n-1$ (where the original hydrocarbon had n carbon atoms. The mechanism of pyrolysis, however, is not clear. Hydrogen and ethylene are found in most cases in the gaseous reaction products.

The temperature at which the reaction rate of pyrolytic reactions becomes appreciable varies with the molecular weight of the hydrocarbons. In the case of the higher molecular weight hydrocarbons with which we are concerned, the rate may be appreciable at 900 F, which is only a few hundred degrees above the end point of distillate fuel oils. Since in certain types of burners the fuel may be subjected to these and higher temperatures cracking may occur to a considerable extent. Available information indicates, however, that high-molecular-weight polymers and carbon are not formed by cracking of paraffinic hydrocarbons.

The pyrolysis of olefinic hydrocarbons may result in polymerization, isomerization, or decomposition. The first and second of these reactions occur at lower temperatures (as low as 600 F). As the temperature is raised decomposition begins and the three reactions occur simultaneously at moderately high temperatures; increase in temperature seems to favor the decomposition reaction. The products formed by pyrolysis of olefins, accordingly, are more numerous and comprise more types than are obtained by pyrolysis of paraffins, e.g., with a higher-molecular-weight olefin ($C_{10}H_{22}$) pyrolysis yielded a complex mixture containing olefinic, paraffinic, and aromatic hydrocarbons. In these experiments unsaturated hydrocarbons of lower molecular weight were formed in considerable quantity at about 1100 F, but aromatic hydrocarbons and tar were formed, together with some carbon, at higher temperatures. Accordingly, the presence of olefins in fuel-oil distillates may give rise to tarry and cokelike deposits if the temperature to which unburned fuel is exposed should become excessively high.

Relatively little information is available on the pyrolysis of naphthenes. It may be postulated that the major portion of the naphthenes in distillate fuel oil is composed of alkyl cyclohexanes, with bicyclic hydrocarbons and alkyl cyclopentanes

present in minor proportions. Theoretically, these naphthenes can react by scission of C—C or C—H bonds in the side chains, scission of C—H in ring (dehydrogenation to corresponding aromatic or cyclo-olefin molecules), or rupture of a C—C bond in the ring and breakdown of the ring structure. Experiments with cyclohexane and its lower homologs have shown that all three types of reaction occur.

As in the case of naphthenes, aromatics may undergo a number of different types of reactions. In addition to the foregoing reactions, scission of a C—C bond between ring and side chain (dealkylation) or simultaneous occurrence of two of these reactions is possible. In cases of C—H scission in the ring or of dealkylation the aromatic radicals formed frequently combine to form "multiple ring compounds," such as diphenyl. Under certain conditions ring fusion occurs with formation of the polynuclear aromatics, such as naphthalene, anthracene, phenanthrene, and the like. It seems probable that cracking of aromatic hydrocarbons in every case (including severe pyrolysis leading to rupture of the ring) yields higher-molecular-weight, low-volatile aromatics. As will be noted later these materials are potential sources of trouble in certain types of oil burners.

Fortunately aromatic hydrocarbons are relatively more stable than the other three types of hydrocarbons, and the rates of cracking in the absence of catalysts does not become appreciable below red heat (about 1300 F). Accordingly it would seem that cracking of aromatic hydrocarbons could be minimized by close control of temperatures.

PROPERTIES OF HYDROCARBONS AFFECTING BURNER DESIGN

Certain physical and physicochemical properties of hydrocarbons and their mixtures are of considerable significance in their utilization as fuels. The most important of these will be defined at this point and their effect will be discussed in more detail in the section on oil-burner performance.

The properties that seem to be of most significance are as follows:

- 1 Volatility. The volatility of a fuel is a measure of the readiness with which it changes from the liquid to the vapor state. Since complex mixtures are used in practice, the volatility is generally defined in terms of the flash point and the distillation range, with particular emphasis on the initial boiling point and ten per cent point in the latter.

- 2 The heat of vaporization is the quantity of heat required to vaporize a given weight of material. This quantity varies for different types of hydrocarbon mixtures with the same volatility, being greater for aromatic than for paraffinic hydrocarbons.

- 3 Viscosity is the resistance of a fluid to flow.

- 4 Limits of inflammability are the limiting percentages of fuel vapor in air defining the region of fuel vapor-air mixtures in which a flame will propagate. There is little variation between fuels of different types, but these limits, in themselves, are of importance.

- 5 Carbon residue, as determined by one of several methods, is a measure of the tendency of the fuel to crack and form, as an ultimate product, petroleum coke.

BURNER DESIGN AND RELATION TO THE COMBUSTION CHARACTERISTICS OF FUELS

The number of devices invented to burn distillate fuel oils is quite large, and the enumeration of them alone would require an excessive amount of time. These devices, however, operate by either one of two methods: They vaporize the fuel, mix it with air, and burn the mixture, usually with the aid of secondary air; such burners are known as vaporizing burners; or they disperse the oil as fine droplets by some mechanical or hydraulic means

and burn the mist of oil droplets after admixture with air; such burners are known as atomizing burners. Although considerable differences exist between individual burners, each can be classified in one or the other of these types. Accordingly the discussion of the relation between burner design and combustion characteristics of fuels will be based on these two types, rather than on different individual kinds of burners.

VAPORIZING BURNERS

From the chronological viewpoint the vaporizing type of oil burner would seem to take precedence over the atomizing type of burner. In this method of burning oil a number of methods have been used to vaporize the oil, namely, from a wick, from a shallow pool, and from a thin layer of oil on a solid surface. In some burners a combination of these methods is used, e.g., in sleeve-type range burners the oil is vaporized from a wick in starting, but from a pool of oil after the metal vaporizing surfaces, sleeves, and the like have been heated to operating temperatures. Novel means of vaporization have been proposed and sometimes employed; this discussion, however, will be limited to the more conventional burners employing vaporization from an oil pool or from a solid surface.

In the conventional pot-type vaporizing burner, Fig. 2, fuel

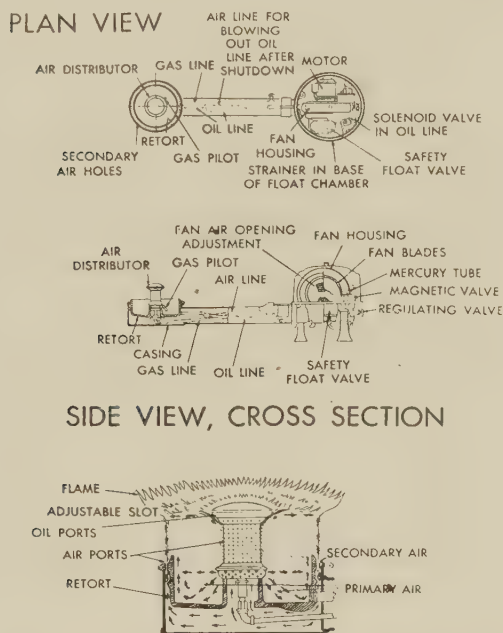


FIG. 2 POT-TYPE MECHANICAL-DRAFT BURNER

enters a shallow concave "dish," either from the center or the side, and forms a pool of oil. By means of a burning piece of paper, cloth, or waste a small amount of oil is caused to vaporize and begin to burn. The heat generated vaporizes additional oil until oil vapors are rising from the entire surface. Air is admitted through openings in a cylindrical shell surrounding the vaporizing chamber. The number of openings varies with the height above the oil surface, being relatively few at the bottom and numerous near the top. As oil vapor begins to form in quantity the mixture in the lower portion of the combustion chamber becomes too rich to burn and the flame rises. Baffles are frequently incorporated in the combustion chamber to improve the mixture. In some burners the air is secured by natural draft, whereas in others the air is supplied by means of a fan.

This seems to be a logical place to note the relation of the

theoretical aspects of combustion to this method of burning oil. The energy necessary for vaporization of the oil, which, as noted previously, varies with fuel type, is supplied by conduction through the metal and by radiation from the burning oil vapors. Inasmuch as the rate of heat transfer by conduction is proportional to area and difference in temperature (transfer by radiation should, it seems, be avoided) there is a definite relation between the rates of vaporization and burning. But the rate of cracking reactions is also a function of the temperature. Fuels of relatively high volatility and low heats of vaporization are accordingly desirable if cracking is to be minimized. Carbon residue is an experimental property of the fuel indicative of its resistance toward cracking.

The oil vapors begin to mix with air a short distance above the oil surface, and since the temperature is considerably greater than that at which slow oxidation occurs these reactions begin as soon as oxygen becomes available. A self-sustaining flame will not appear until the fuel vapors have been diluted with at least thirteen volumes of air, i.e., until the mixture becomes leaner than that defined by the upper limit of inflammability. In this volume containing mixtures too rich to support a flame oxidation reactions occur, with formation of aldehydes, peroxides, and other oxidation and hydrocarbon degradation products. In general this is not deleterious, but in designs in which these vapors may strike a cold surface, deposition of gumlike materials, which may be the precursors of tar and coke, may occur. Since rates are the important consideration, much of the problem of design must be concerned with the time factor, both in regard to vaporization and mixing of fuel vapors and air.

Vaporizing burners are sometimes low in efficiency and in general are subject to soot and carbon formation. From the preceding discussion the reasons for the difficulties encountered in practice in obtaining high efficiency and completeness of combustion are rather easily discernible. Briefly they are as follows:

- 1 There is, in general, inadequate provision for regulation of air flow. Consequently a gross excess of air may exist at low fire and a deficiency at high fire.

- 2 The means of mixing fuel vapors and air are frequently primitive and relatively inefficient.

- 3 The temperatures of liquid fuel and very rich oil-air vapors are not controlled. Consequently cracking reactions may be appreciable in both liquid and vapor phases. As indicated previously this leads to formation of high-molecular-weight low-volatility hydrocarbons, particularly when fuels containing aromatics are being burned, which may condense on the sides of the combustion chamber and by continued cracking ultimately yield tars and coke.

- 4 Frequently the oil is exposed to radiation from the flame. It is conceivable that photochemical reactions occur, which aggravate the tendency toward soot and carbon formation.

- 5 Frequently flames impinge on relatively cool surfaces. Since, in general, the flames are luminous and the mixture is frequently rich at the points of impingement the cooling of the flame results in deposition of finely divided carbon, i.e., soot, on these surfaces.

In designing a vaporizing burner the foregoing points should be borne in mind. The temperatures of liquid and vapors of fuel oil should preferably be kept well below the cracking temperature, yet provision should be made to supply a sufficient amount of heat to vaporize the required quantity of fuel. Fuel vapors and air should be formed into a homogeneous mixture too rich to support combustion, which is then burned, preferably under turbulent conditions, in the presence of sufficient secondary air to complete combustion. Impingement of flames on

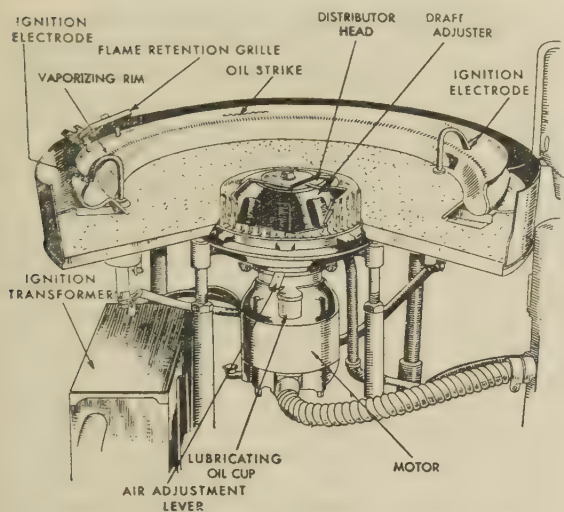


FIG. 3 WALL FLAME-TYPE VAPORIZING BURNER

cool surfaces is to be avoided, and irradiation of liquid oil or vapor-air mixtures in the primary zone should be prevented (6).

A vaporizing pot-type burner is now in production which operates at relatively high vaporizing temperatures. Although the authors have no information regarding the operating characteristics of this burner, it is plausible that such a burner could perform satisfactorily if special care were taken to prevent vapor-air mixtures from contacting any cool surfaces and means were provided for very good fuel vapor-air mixing. The time factor is very important in regard to carbon and soot formation, and it is easily conceivable that slow vaporization at a relatively low temperature could result in a greater degree of cracking than very rapid vaporization at a temperature several hundred degrees higher despite the much greater reaction rate at the higher temperature.

A vaporizing burner design in which most of the foregoing difficulties have been avoided is the wall-flame-type vaporizing burner, shown diagrammatically in Fig. 3. In this burner fuel is thrown by centrifugal force against a hot steel or ceramic surface on which it vaporizes. Air from a fan mixes with the oil vapors and the mixture burns above a series of grilles, or flame retainers, placed above and slightly back of the vaporizing surfaces. Inasmuch as the quantities of air and fuel can be regulated and the distance of the flame retainers from the vaporizing surface can be varied (i.e., the mixing time of fuel vapor and air can be adjusted) this type of burner is capable of giving efficient performance on various types of fuel (7). Ignition is the operation most likely to give difficulties with this type of burner.

ATOMIZING BURNERS

The second method of burning fuel oil consists of dispersing the oil in fine droplets, admixing these with air, and burning the mixture in a suitable combustion chamber. High-pressure mechanical atomization, air atomization, and centrifuging from a rotating "cup" are the most commonly used methods of dispersing the oil. Burners employing these methods of atomization are illustrated in Figs. 4 to 6. Diverse means are also employed to form the mixture of fuel droplets and air. Varying degrees of combustion efficiency have been attained, dependent primarily on the degree to which good mixing of air and fuel has been attained.

The processes involved in burning fuel oils with this type of burner would seem to differ radically from those encountered in vaporizing burners. On a macro scale this is true, but from the micro viewpoint certain similarities can be ascertained. Each

droplet can be regarded as a vaporizing unit. Surrounding each drop is a volume of fuel vapor, very rich at the oil surface and becoming more dilute as the distance from the center increases. At a given radial distance the mixture is sufficiently lean to support combustion and there a flame front forms. Fuel vapor diffuses into the flame front from one side, air from the other, and the process continues until the droplet is consumed. Cracking and slow oxidation reactions may occur, accordingly, although the latter are probably of less significance than in vaporizing burners. Fuel properties such as volatility, heats of vaporization, carbon residue, and limits of inflammability are of less significance, it would seem, than in the vaporizing burner.

The operation of atomizing burners is dependent principally on two factors, namely, the degree of atomization and the formation of air-fuel mixture. The relation of these factors to the combustion processes and fuel properties are discussed at some length.

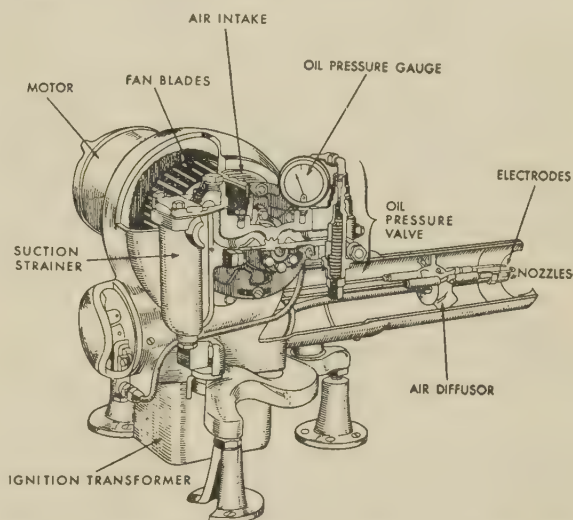


FIG. 4 HIGH-PRESSURE MECHANICAL-ATOMIZING GUN-TYPE OIL BURNER

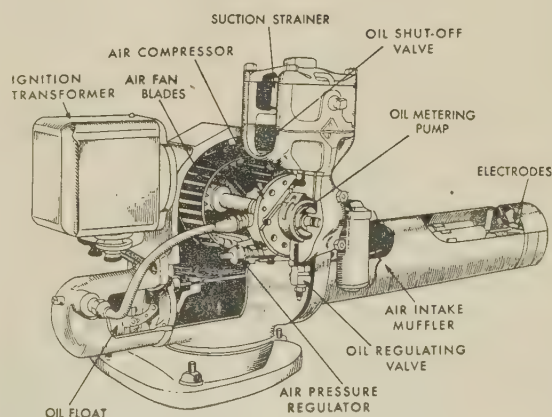
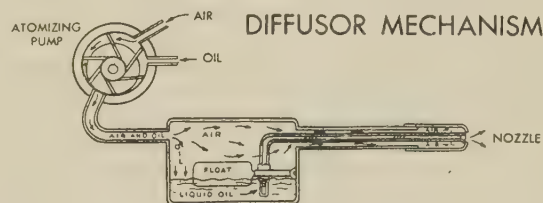


FIG. 5 LOW-PRESSURE AIR-ATOMIZING GUN-TYPE OIL BURNER

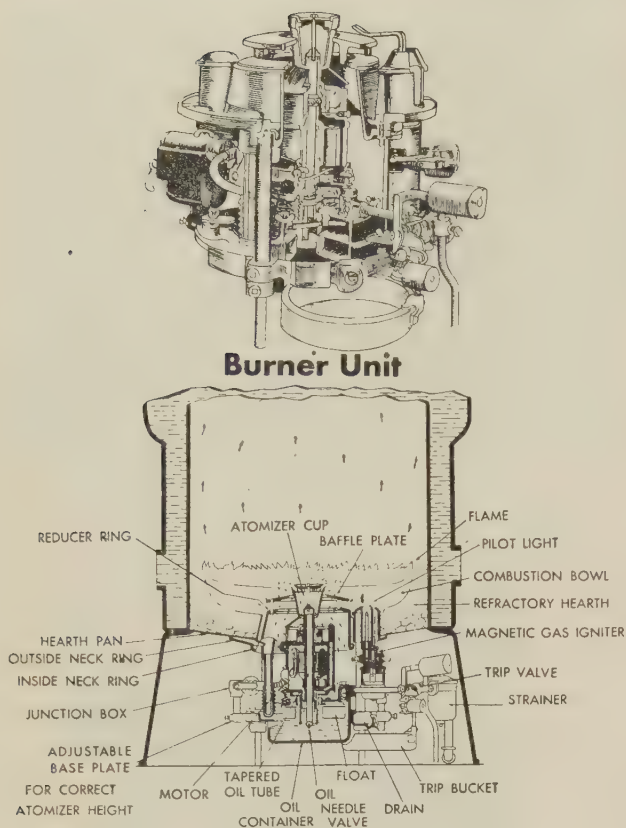


FIG. 6 VERTICAL ROTARY ATOMIZING OIL BURNER

The process of atomization has been studied rather intensively, particularly when accomplished by means of pressure. It has been shown that the quality of the spray (as judged by the spray angle and burning characteristics in a given installation) is markedly affected by changes in viscosity of the oil (8). Presumably the particle-size distribution is also affected. It seems logical to assume that the rate of combustion is proportional to the total area of oil exposed, hence is dependent on the particle-size distribution. It is well known that the vapor pressure of a liquid increases as the diameter of the droplet decreases; as the mean diameter decreases, too, the area available for heat transfer increases, thus increasing the rate of vaporization. These factors, namely, the mean particle size, particle-size distribution, spray form, and spray angle, govern the volume of space required for combustion. Since the droplets are in a high-temperature zone and are presumably surrounded by an envelope of flame in most cases, the heat of vaporization and volatility are properties of minor importance.

In domestic burners, which operate at relatively low combustion rates, the major problem is to secure an intimate mixture of fuel droplets and air. A number of means have been utilized to achieve this end, with varying degrees of success. In the simplest designed burners no special provision for forming the mixture is made, and the burners, consequently, are relatively inefficient. Cones, swirl vanes, disks, and other devices have been introduced into the burner tube to direct the air stream and/or increase the turbulence of the air flow and thus increase the effectiveness of mixing of fuel and air. In other cases the combustion chamber has been shaped to promote certain flow patterns in the gases and thus effect better mixing of air and fuel. Even with these devices the ideal mixture of air and fuel is probably never obtained and satisfactory mixtures but infrequently.

As in the case of the vaporizing process the liquid fuel and fuel

vapor are subjected to high temperatures and relatively intense radiation. The cracking reactions discussed in a preceding section occur at an appreciable rate, and since the temperatures are quite high carbon may be formed. If sufficient oxygen is present this carbon will be consumed; if not, soot and smoke result. If in addition the fuel spray or flame impinges on the surface of the combustion chamber, cooling occurs and smoke and carbon are formed due to retardation of the combustion process. If, in addition, some of the fuel remains on the wall in the liquid state it undergoes intense cracking and ultimately builds up cokelike deposits. This aggravates the condition, so that large deposits may be built up in a relatively short time under these conditions. To prevent smoke and carbon formation it has been a practice to use an excess of air, but this results in an over-all drop in the efficiency of the burner.

An ideal atomizing burner, it would seem, would produce a mist of oil particles of relatively uniform size, with the spray angle and spray distributions to conform with the shape of the air-flow pattern. A partial mixing of air and fuel vapor should occur before passage into the flame front, which should be stable in form and position. Additional air to complete combustion should be introduced either by the burner fan or from the surrounding atmosphere, in the latter case the flame itself drawing in the secondary air. Excess air, above a small proportion deemed necessary to insure complete combustion, should be avoided. Possibility of impingement on combustion-chamber walls or portions of the burner should be eliminated.

CONCLUSIONS

- 1 The slow oxidation of the gaseous and lower-boiling liquid hydrocarbons has been studied intensively and mechanisms to explain the experimental results have been proposed.

- 2 Hydrocarbons in the fuel-oil range when admixed with air and heated to temperatures lower than the ignition temperature undergo oxidation. The mechanisms of oxidation are analogous to those postulated for the lighter hydrocarbons.

- 3 Hydrocarbon molecules when heated, alone or in the presence of air, undergo "cracking" reactions, involving scission, isomerization, and polymerization.

- 4 The mechanisms of oxidation in the flame are probably different from those operative at lower temperatures. Our present knowledge of reactions occurring in the flame are derived largely from spectroscopic studies.

- 5 The principles relating to hydrocarbon oxidation are applied to the two more common methods of burning distillate fuels, i.e., in vaporizing or atomizing burners, and specific recommendations are made with regard to their application in burner design.

BIBLIOGRAPHY

- 1 "Combustion, Flames, and Explosions of Gases," by B. Lewis and G. von Elbe, Cambridge University Press, London, England; The Macmillan Company, New York, N. Y., 1938, chapter 1.
- 2 *Ibid.*, chapter 4.
- 3 *Ibid.*, chapter 5.
- 4 "The Emission Spectrum of Hydrocarbon Flames," by E. C. W. Smith, Proceedings of the Royal Society of London, series A, vol. 174, 1940, pp. 110-125.
- 5 "Reactions of Pure Hydrocarbons," by G. Egloff, Reinhold Publishing Corporation, New York, N. Y., 1937 (American Chemical Society Monograph Series No. 73).
- 6 "Combustion of Catalytically Cracked Distillates and New Burner Design Widening Markets for These New Fuels," by S. R. Cauley and H. R. Linden, *Oil and Gas Journal*, vol. 45, No. 14, 1946, pp. 80-82, 86, 91.
- 7 "Fuel Oils Have Changed," by W. A. Sullivan, *Scientific American*, vol. 175, 1946, pp. 66-68.
- 8 "Atomization of Oil by Small Pressure-Atomizing Nozzles," by E. B. Glendenning, A. R. Black, L. H. Ventres, and W. A. Sullivan, *Trans. A.S.M.E.*, vol. 61, 1939, pp. 373-381.

Discussion

O. F. CAMPBELL.⁴ This paper points out the intricate and complex combustion problems which should be known by the designer of domestic oil-burner equipment. Lewis and von Elbe's postulated mechanism of hydrocarbon oxidation by a material chain process throws further light on the complex problem. Mr. Reed of Tulsa has indicated that under certain conditions the aldehydes are formed in the combustion process and gives a quick method of aldehyde detection.

However, regardless of theories, we must face the facts that some domestic fuels burn better than others in the same burner, and oils that burn equally well in the same burner do not burn the same in burners of different types. These facts are easily understood when one considers the various sources and grades of crude, the different oil-manufacturing processes for producing domestic fuels, and the different types of oil burners available on the market or in use.

In regard to the different burning characteristics of different fuel oils, a simple illustration should suffice. An oil molecule can be compared with a stick of wood with shavings attached. The carbon content of the oil molecule may be considered the main part of the stick and the hydrogen content may be considered the shavings. If the shavings are uniformly spaced and ignited, the stick will burn up completely at a uniform rate without leaving any unburned stick or carbon deposits. If the shavings are of the same number but not uniformly spaced, the stick will not burn uniformly and carbon deposits are likely. If the shavings are fewer in number and not uniformly spaced, poor burning and carbon deposits are assured.

With numerous types of oil burners, it can be seen readily that the mixing of air and fuel cannot be identical and, in most cases, it is poor. It is the writer's opinion that any fuel can be burned satisfactorily if there is a high enough temperature, enough time for combustion, and an adequate air-fuel mixture exists. Normally, there is enough time and temperature, but it is the rare occasion when adequate mixing occurs.

R. N. ST. JOHN.⁵ The authors mention several shortcomings of vaporizing-type burners which are obviously more pronounced when cracked oils are used in place of straight run oils for which most existing pot-type burners were designed.

An oil-burner installation is considered by the writer to be made up of several essential elements including the following:

- 1 Fuel-control mechanism.
- 2 Burner.
- 3 Combustion chamber and heat-recovery surfaces.
- 4 Blower or chimney or other provision for securing proper burner draft and disposing of flue gases.

Should any of these elements prove inadequate in design or performance, the over-all effectiveness of the installation must suffer.

Improvements in design and performance of vaporizing-type burners may have been retarded by our self-imposed restrictions calling for the following qualities or characteristics:

- 1 Extreme simplicity of construction and operation.
- 2 Dependence upon natural draft as produced by short chimneys in small homes where electric power is not available.
- 3 Maximum heat output from small compact burners and heat-recovery surfaces which are designed for minimum use of

space in the home, and ease of cleanout in cases of deficient draft.

Even with these restrictions much progress has been made and complete packaged heating units have shown over-all efficiencies which compare favorably with those obtainable with other types of burners.

Even at low firing rates, over-all efficiencies do not drop as rapidly as might be expected. The flue-gas temperatures and resulting flue-gas losses are sharply reduced at the lower firing rates, and, as chimneys cool off, the effective draft is also reduced so that over-all efficiencies at low fire are often within 5 to 7 per cent of high fire efficiencies. These, in turn, are often limited more by lack of heat-recovery surface than by actual burner-performance limitations.

Soot and petroleum coke form in most pot-type burners during the first few hours or days of operation, but the rate of such residue formation drops off rapidly with continued use. With proper draft there generally is some firing rate at which accumulated carbon or coke deposits ignite and burn out more or less completely at sufficiently short intervals to permit extended use of the burner without manual cleanout service.

A definite trend toward the production of cracked fuel oils in recent months has resulted in much active research and experimental work on the part of manufacturers of oil burners of the vaporizing type, and with the help of the petroleum industry we believe that definite improvements in the design and performance of such burners should be forthcoming in the not too distant future.

J. C. REID.⁶ In general, the background of hydrocarbon oxidation has been well presented considering the necessarily limited length of the paper. Extension of our present knowledge in this direction should prove helpful in expanding the fundamental treatment of combustion properties. However, the correlation of these reactions with burner performance and design is far from being established at the present time, and it would appear that several intermediate steps are needed before such correlation can be made effectively. Solution of these steps, mechanical and physical problems, as recognized in this paper, would result in gratifying improvements in burner performance without the necessity of detailed knowledge of specific oxidation reactions.

The effect of the fuel-air ratio, intimacy of mixing, etc., should be emphasized strongly in relation to the type of flame. Optimum operating conditions should give substantially the same flame regardless of the chemical nature of the fuel.

The "competing reactions" cited conceivably can take place in the vapor as well as in the liquid phase. Speedy vaporization in itself, therefore, will not completely overcome the difficulties engendered by these reactions.

The chemical nature of the fuel will also influence the design of an ideal burner, although any installation which accomplishes the aims stated, more or less automatically would take care of fuel variations.

In discussing vaporizing burners, it is believed that the chemical composition of the fuel is especially important. It would also appear that in any explanation of the low efficiency of these burners, the means of mixing fuel and air are intimately related to the regulation of air flow. The points affecting design are well taken. The fuel will define temperature limitations to some extent and thus will be a factor in design.

Atomizing burners are less sensitive to these factors. However, the same reasoning applies. The difference lies in the mechanics of the solution of the problems involved. It should be borne in mind that large excesses of air can cause cooling of the

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⁶ Atlantic Refining Company, Philadelphia, Pa.

flame and thus cause carbon and soot troubles. Secondary air is important and better mixing of this air in the combustion process should be sought in improving burner design.

As developed in this paper, it would seem that "cracking" reactions, at the present state of our knowledge, are extremely important factors in burner design. In this connection it may be reiterated that understanding of oxidation reactions needs considerable expansion to be of real assistance. Such understanding must be accompanied by increased knowledge of the interrelation of oxidation and pyrolysis as encountered in installations for burning fuel oils. The recognition of some of these points as presented in the paper under discussion should aid in improving design and accomplishing many of the ultimate aims without awaiting the detailed theoretical study of specific flame reactions.

AUTHORS' CLOSURE

The comments of O. F. Campbell with regard to differences in performance of the same fuel in different burners, or of different fuels in the same burner, indicate complexity of the problem from the practical point of view. The writer feels that the figure of the stick of wood with shavings attached somewhat over-simplifies the problem, however. To extend the figure one step further, the ease with which the wood is ignited and burned depends not only on the arrangement and spacing of the shavings, but also on the type of wood of which the stick is composed, i.e., whether hardwood or softwood. Likewise, the arrangement of the carbon

atoms in the molecule is of as much significance as the spatial relations of the carbon and hydrogen atoms in the combustion characteristics of a fuel.

R. N. St. John has presented a very concise analysis of the component parts of the oil-burner mechanisms and of the restrictions imposed upon manufacturers of vaporizing-type burners in the improvement in design and performance of this type of burner. It is recognized that these burners, within the delineated limitations perform relatively satisfactorily for any given type of fuel oil. On the other hand, this type of burner is relatively inflexible in situations wherein the ratio of straight run to cracked components in the fuel fluctuates from time to time. It is encouraging to note that research programs have been initiated to overcome the poorer performance of this type burner with cracked fuels; it should be noted, however, that some of the self-imposed limitations which the vaporizing-type-burner manufacturers have set up may have to be modified or abandoned if a satisfactory solution of the problem of flexibility is to be attained.

The emphasis placed on "competing reactions" by J. C. Reid is shared by the authors. More information regarding "cracking" reactions, both in the liquid and the vapor phase, is highly desirable. Particularly needed are data on the "cracking" of hydrocarbons in the presence of oxygen or air. It is to be hoped that a paper dealing with the relation of these reactions to oil-burner design and operation can be presented in the near future.

Furnaces for By-Product Fuels

By OTTO DE LORENZI,¹ NEW YORK, N. Y.

Present-day costs of standard fuels such as coal, oil, and gas have advanced to a high level and concepts must now be revised regarding so-called refuse fuels. Their value as low-cost substitute fuels, for steam generation, will depend largely upon the manner in which they are prepared, and upon the use of suitable furnace designs. A number of furnace designs have been developed to handle by-product fuels from oil-refinery operations, and the manufacture of steel, coke, lumber, pulp, and sugar. Others are still in various experimental stages and many improvements in methods of drying, feeding, and burning fuels will be presently available. The characteristics of several of these fuels are discussed, and actual furnace designs illustrate the manner in which they are fired to provide process steam and save fuel dollars.

BY-PRODUCT fuels may be classified broadly as those which constitute some of the by-products of various manufacturing processes. Their value as low-cost substitute fuels, for steam generation, will depend largely upon the manner in which they are prepared and on the use of suitable furnace designs. Perhaps the most widely available of these fuels result from oil-refinery operation and the manufacture of steel, coke, lumber, pulp, and sugar. Of course there are many others but their availability is so limited that no attempt will be made to discuss them at this time. Typical analyses for some of these fuels are shown in Table 1.

still gas are names applied to some of the gaseous by-products.

Asphaltic Pitch. When the distillation of oil is stopped somewhat early, there remains a residue which is solid at room temperature, but fluid at still temperature. This residue is asphaltic pitch, with a melting point of about 125 F, and it is usually pumped direct from process to burner at a temperature of 300 F to 600 F. Under these temperature conditions it is a fluid which is readily atomized, and which can be burned without difficulty. Its properties are somewhat similar to the heavier grades of bunker C oil and therefore furnace-design limitations are practically the same.

Petroleum Coke. Petroleum coke is the solid residue remaining after cracking or carrying the distillation of crude oil sufficiently far. Its characteristics depend upon the process used. Volatile-matter content varies from 4 to 12 per cent, and the sulphur and ash also vary widely. Ash-fusion temperature may be as low as 2000 F. Grindability ranges from that of extremely hard and abrasive coke, similar to metallurgical coke, down to one that can be easily pulverized with low power consumption and low mill maintenance. This pulverized fuel is readily burned in water-cooled furnaces, provided moderate heat-liberation rates, between 15,000 and 22,000 Btu per cu ft per hr, are used.

Acid Sludge. The characteristics of a sludge are governed by those of the crude oil used, and the manner in which it is processed. Much of the suspended solids may be carbonaceous, in the form of small particles of petroleum coke. Continuous agitation and recirculation at velocities high enough to prevent

TABLE 1 TYPICAL ANALYSES OF SOME SPECIAL FUELS

Constituents	Per cent by weight						Per cent by volume saturated with moisture at 62 F and 30 in. Hg	
	Bunker C oil	Acid sludge	Coke breeze	Red-wood chips	Hogged hem-lock	Bagasse	Refinery gas	Blast-furnace gas
Moisture.....	Trace	17.5	5.0	50.4	53.7	43.7
Carbon.....	84.0	63.5	85.7	53.5	50.4	43.1
Hydrogen.....	12.5	8.1	0.7	5.9	5.8	6.2	..	1.8
Oxygen.....	1.2	..	0.5	40.3	41.4	47.9	1.2	..
Nitrogen.....	1.7	..	1.3	0.1	0.1	..	7.7	56.5
Sulphur.....	0.4	3.2	1.0	..	0.1
Ash.....	0.2	4.3	10.8	0.2	2.2	2.9
Oxygen plus nitrogen.....	..	3.4
Carbon dioxide.....	8.7
Carbon monoxide.....	0.3	32.8
Methane.....	36.3	0.2
Ethane.....	22.9	..
Propane.....	26.8	..
Butane.....	4.3	..
Pentane.....	0.5	..
Higher heating value:								
Btu per lb (dry).....	18445	14341	12690	9220	8620	7985	1590	111
Btu per cu ft.....

OIL REFINERY BY-PRODUCT FUELS

By-products from oil-refinery operations consist of a wide variety of refuse fuels. There are solids, such as asphaltic pitch and petroleum coke. The liquids or sludges are often of high specific gravity and contain variable amounts of solid matter in suspension. Refinery gas, blended refinery gas, yard gas, and

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settling out are necessary to avoid plugging of fuel lines.

Perhaps the most widely available and yet most troublesome, because of its frequently varying characteristics, is acid sludge. Its gravity may range between 5 and 14 API, and its viscosity is indeterminate. It contains varying quantities of weak sulphuric acid which may reach as high as 40 per cent and this, together with the suspended carbonaceous material and flux, which must be added in variable amounts to make the sludge flow, causes the heating value to vary between 8000 and 17,500 Btu per lb. Due to the suspended solid matter, it is necessary to use relatively large orifices in the atomizer. As a result, atomization is coarse, and ignition is not always stable. Refinery practice is to burn the sludge as a supplementary fuel to gas or oil. Each

steam-generating unit is provided with several combination-type burners, some of which operate, with gas or oil, to maintain ignition and reasonable capacity during periods when sludge supply is of erratic quality. The products of combustion, resulting from acid-sludge burning, carry a large quantity of water vapor. Gas temperature, at the outlet of heat-recovery equipment, must be well above the dew point if moisture deposit and resulting corrosion of the metallic surface is to be avoided. Maximum furnace-liberation rate should not exceed 25,000 to 30,000 Btu per cu ft per hr.

Refinery Gas. The heating value of refinery gas is higher than that of natural gas, owing to the larger percentage of heavier hydrocarbons present. Its composition is variable as a result of differences in the characteristics of the oil refined and the extent of cracking to which the oil has been subjected in order to extract gasoline. There are also present some illuminants, or unsaturated hydrocarbons from the cracking operations.

For use in steam-generating units, the gas from several types of operations is mixed, or blended. The characteristic flame from this blended gas, except for a distinct, clear, blue zone in the area of the burner throat, is colorless and extremely short. At times, however, the gas may be wet and contain some of the lighter phases of gasoline in the forms of mist or vapor; when present these burn with small, intermittent, white, flashing flames. These gasolines are undesirable because of their comparatively slow-burning characteristics, which result in continuous or secondary combustion that may extend through a large portion of the boiler. As a result, efficiency is lowered because of increased losses resulting from higher boiler-exit temperature; and increased maintenance follows, because baffle and superheater supports are subjected to abnormally high temperature.

Burners for refinery gas are similar to those used for natural-gas firing, and are also arranged for using auxiliary liquid fuels. The gas ring or tube should be readily replaceable as there may be a tendency for orifice plugging from carbon accumulations when burning contaminated or wet gas.

Furnace-design limitations are the same as those for natural-gas firing. Fully water-cooled walls may be employed and maximum continuous heat-liberation rate should not exceed 20,000 to 30,000 Btu per cu ft per hr.

The steam-generating unit shown in Fig. 1 is one of several installed for a large eastern oil refinery. It is designed for a continuous capacity of 125,000 lb of steam per hr at 800 psi and 750 F, with initial operation at 450 psi. The fuels are bunker C oil, refinery gas, soda tar, and acid sludge, with provision for future pulverized-coal firing. Four horizontal turbulent-type burners, arranged to handle any of the fuels, are used. The units are often operated by burning three of these fuels simultaneously. The most frequently used combination is oil, soda tar, and refinery gas.

The units have regenerative-type air heaters. The furnaces are water-cooled and designed for a continuous heat-liberation rate of 25,000 Btu per hr with oil at a corresponding efficiency of 84.3 per cent. With soda tar, the efficiency is 85.6 per cent, and with acid sludge 84.8 per cent.

STEEL MILL BY-PRODUCT FUELS

The two principal by-product fuels from steel-mill operation are blast-furnace gas and coke breeze.

Blast-Furnace Gas. Blast-furnace gas results from the various reactions occurring in the different zones of the blast furnace. It contains relatively high percentages of carbon monoxide along with carbon dioxide, nitrogen, and water vapor. It is a lean gas, having a heating value which varies between 90 and 110 Btu per cu ft and is dependent upon the quality of coke used, the speed of combustion, the ore treated, and many other factors. As this

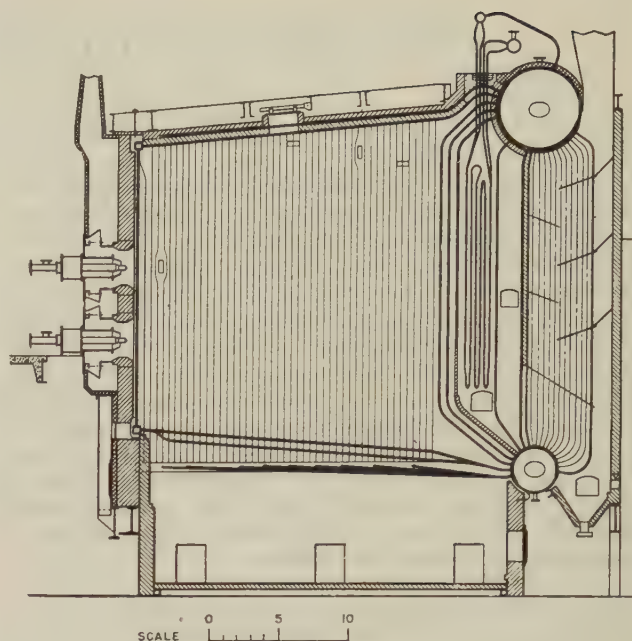


FIG. 1 ONE OF SEVERAL STEAM-GENERATING UNITS, INSTALLED FOR A LARGE EASTERN OIL REFINERY, DESIGNED TO BURN REFINERY-REFUSE FUELS AND PULVERIZED COAL
(Capacity: 125,000 lb steam per hr at 800 psi and 750 F.)

gas leaves the top of the blast furnace it is hot and contains considerable dust having a high iron-oxide content. Much of the dust is removed and then sintered for return to the blast furnace as a part of the ore charge.

The extent to which blast-furnace gas should be cleaned, for use as a boiler fuel, can only be determined after a careful economic analysis, for each installation, of the many factors involved. The decision whether to burn a dirty, partially cleaned, or a clean gas will have considerable influence on the selection of burners and type of furnace to be employed. The use of hot, dirty gas is feasible even in all-refractory boiler furnaces. On the other hand, in many instances it has been found profitable to clean the gas thoroughly, because furnace outage for cleaning and refractory maintenance are reduced to a minimum.

Blast-furnace gas is slow to ignite as it contains a high percentage of noncombustible gas. Therefore it is important that the burners produce rapid and intimate mixing of this lean gas with the optimum amount of air to assure complete combustion, and at the same time maintain stabilized ignition. Preheating the combustion air will serve to accelerate ignition, and permit the use of completely water-cooled furnace walls with either washed or unwashed gas. The furnace hearth should be shaded by a water-screen so that any accumulation of dust in that area will not be sintered and thus become difficult to remove. Maximum continuous heat-liberation rates should not exceed 20,000 to 23,000 Btu per cu ft per hr.

One of four large tangentially fired units installed in a Midwestern steel mill is shown in Fig. 2. The burners located in the corners of a completely water-cooled furnace, are arranged for blast-furnace gas, and coke-oven-gas firing. Each burner bank is composed of five fuel nozzles, two for blast-furnace gas, two for coke-oven gas, and one for a pilot light. The maximum continuous capacity, when burning blast-furnace gas only, is 200,000 lb steam per hr at 325 psi and 700 F, at a corresponding heat-liberation rate of 23,000 Btu per cu ft, and an efficiency of 79.4 per cent. Combustion air is supplied to the burners at room temperature and, even though the furnace is fully water-cooled,

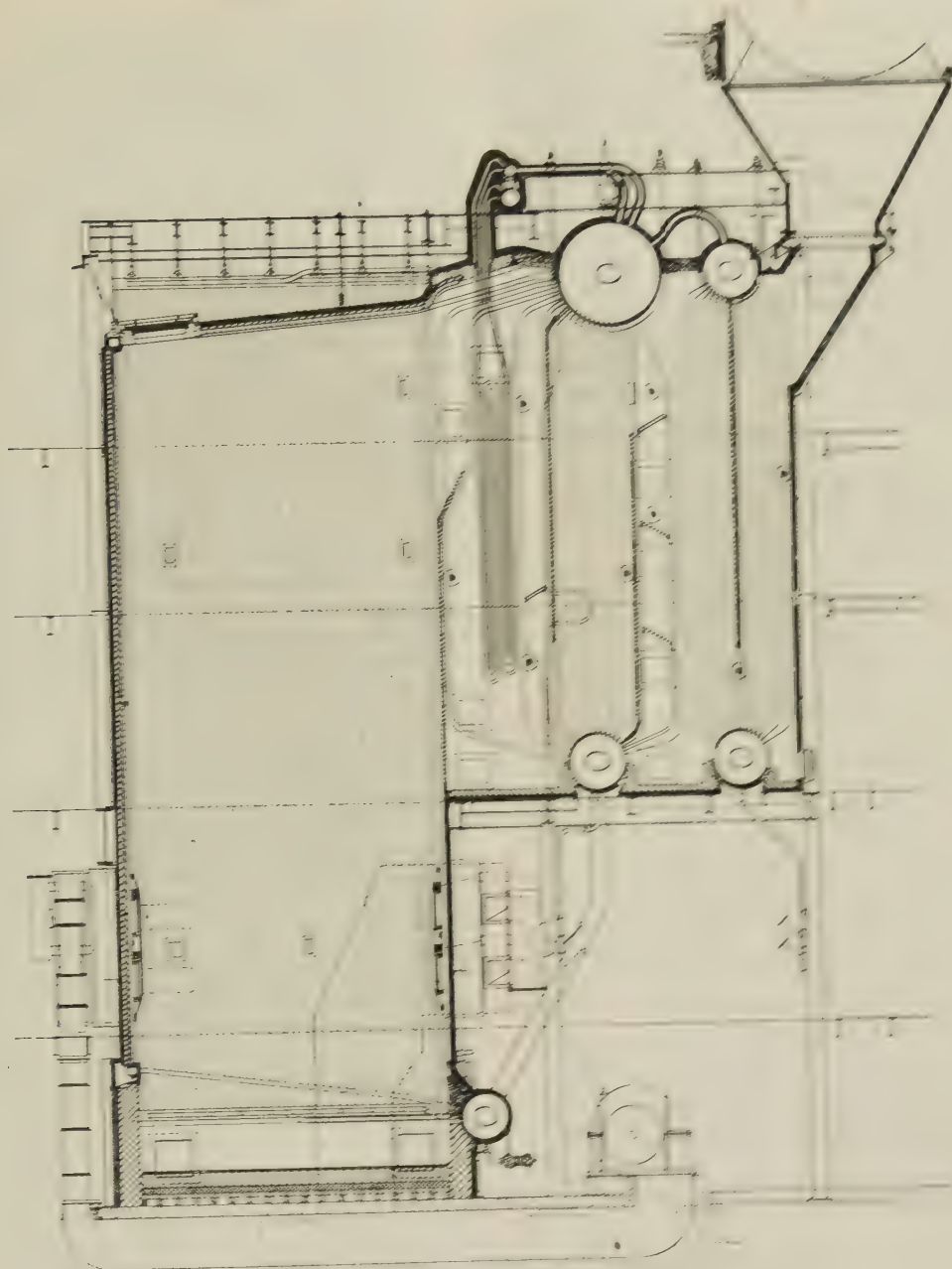


FIG. 2 ONE OF FOUR LARGE STEAM-GENERATING UNITS WITH TANGENTIAL BURNERS, ADAPTED FOR BURNING METALLURGICAL AND COKE-OVEN GAS
(Capacity: 200,000 lb steam per hr at 325 psi and 550 F)

the use of tangential firing assures prompt ignition with rapid and complete combustion when using primary washed gas. The use of a two-drum integral economizer serves to reduce the exit-gas temperature to 500 F, which, together with the low excess-air requirements, makes possible the high efficiency reported.

Coke Breeze. In the manufacture of metallurgical and domestic coke there is always some coke too small even for domestic heating. This by-product is generally known as "coke breeze." There is as yet no accepted standard for the sizing of breeze, even in those plants where it is produced. For that reason, no fixed design of furnace has been developed for burning this fuel. Furthermore, it is most difficult to ignite because it has a sponge-like structure and a volatile-matter content which lies in the low

range of 1 to 7 per cent. By far the largest quantity of waste comes from the manufacture of metallurgical coke and has a maximum volatile-matter content of 2 per cent.

Since breeze is the principal factor in burning coke waste successfully, the traveling or chain-grate burner is the most suitable type to use. With these the fuel is progressively fed to the furnace in a thin, magnified, slow-moving layer which is exposed to radiant heat from unitarily placed burners. Ignition is prompt, stable, and penetrates the fuel bed rapidly. The grate is under such close observation that the operator may regulate the quantity and form of the fuel bed at any time.

In designing furnaces for burning coke waste, the most important factor is the volatile-matter content of the waste, which varies

fuel is $\frac{1}{4}$ in. to 1 in., the front-arch design is chosen because it has a heat-stabilizing effect on ignition of the fuel. If the maximum sizing of breeze is $\frac{1}{2}$ in. to $\frac{3}{4}$ in., the rear-arch design is employed. Minimum sizing, or undersize, of breeze is also of considerable importance because of its effect on carbon loss in ash pit and fly ash. Coke breeze is much lighter than coal and the fine particles lifted from a fuel bed are easily carried out of the furnace with the gas. It can be said, in general, that best results are obtained with coke breeze if it is screened to pass through a $\frac{1}{2}$ -in. or $\frac{3}{4}$ -in. round-hole screen, and contains not less than 20 per cent or not more than 30 per cent undersize of $\frac{1}{2}$ in. or smaller.

Frequently, steam-generating units whose principal fuel is coke breeze are also arranged for supplementary firing of either blast-furnace gas or coke-oven gas. A successfully operating unit of this type is shown in Fig. 3. A maximum capacity of

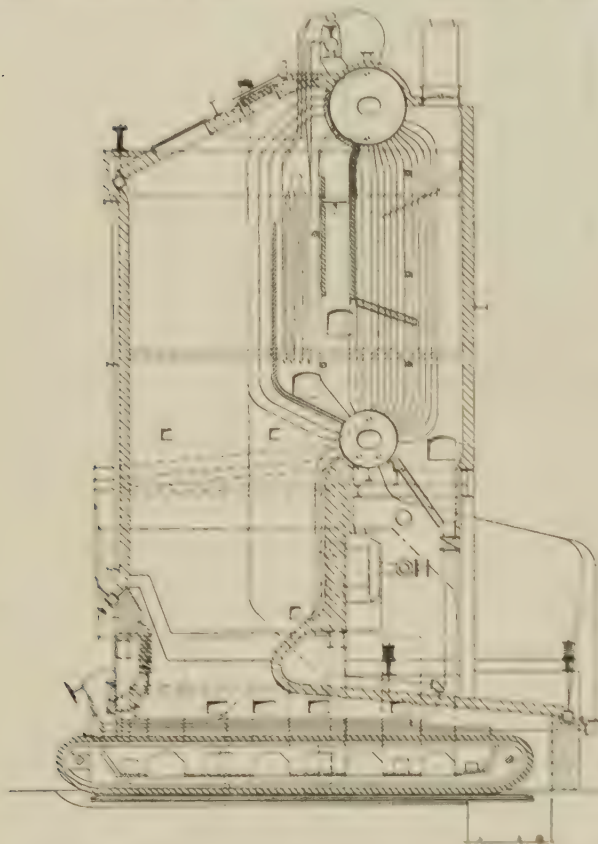


FIG. 3 STEAM-GENERATING UNIT DESIGNED TO BURN EITHER COKE BREEZE OR COKE-OVEN GAS
(Capacity: 45,000 lb steam per hr at 415 psi and 650 F.)

45,000 lb of steam per hr at 415 psi and 650 F is developed with either coke breeze or coke-oven gas. At this capacity, the heat-liberation rate in the secondary furnace for gas firing is 25,000 Btu per cu ft. The stoker area was selected for the conservative burning rate of 30 lb of breeze per sq ft per hr. The rear arch and the front wall of the secondary combustion chamber are water-cooled. Clinker chills are provided along the side walls at the stoker grate line.

BY-PRODUCT FUELS FROM LUMBER MANUFACTURE

In the vast forest-covered areas of our Northwest are located a majority of the lumber industries' largest sawmills. They provide large quantities of wood waste for steam-generating purposes. Some of this wood is burned at the mill to supply needed power, while much of the remainder is sold for use in utility and

industrial power plants, located within economical shipping distance, to replace more costly coal or oil.

Hog Fuel. In the manufacture of lumber, the amount of material removed from the log to produce sound lumber is approximately as follows: 18 per cent in the form of slabs, edging, and trimming; 10 per cent as bark; and 20 per cent as sawdust and shavings. While the total waste material will usually average around 50 per cent, distribution of different types of waste may vary widely from the approximation given. The mills frequently use the sawdust, or a mixture of sawdust and shavings, for steam-production purposes because they can be burned without further processing. The remainder of these so-called waste products require size reduction in a hog to facilitate feeding, rapid combustion, transportation, and storage. These newly sized products, together with varying percentages of sawdust and shavings, constitute "hog fuel." The moisture content of this fuel may vary from approximately 40 to 55 per cent. In addition, it may range from salt-water hemlock to fresh-water fir. These factors of moisture, wood species, and whether fresh- or salt-water-borne logs were used, are of considerable importance in providing adequate and suitable furnace designs.

Hog fuel is bulky and the basis for purchase and sale is volume rather than weight. The accepted standard is the "unit," equivalent to 200 cu ft of hog fuel as measured in the containing vehicle of transportation, without packing. The weight of a unit will vary from 1700 lb to 2300 lb of dry wood, depending upon the species, the moisture content, and the amount of shavings and sawdust present in the mixture.

The process of combustion with hog fuel, because of high moisture and volatile-matter content, consists in three consecutive and somewhat overlapping stages, i.e., preliminary drying or evaporation of the moisture, distillation and burning of the volatile matter, and burning of residual fixed carbon.

The design of a suitable furnace for hog-fuel firing must take into consideration the manner in which the combustion process is carried on, and at the same time make due allowance for possible wide variation in wood species, size, and moisture content.

A two-stage furnace, comprising a Dutch oven for the drying and gasification of the fuel, and a secondary furnace in which combustion of the gaseous products is completed, provides a relatively simple yet effective arrangement.

Hog-fuel firing is a matter of surface combustion, and the use of refractory arches and walls in the primary or Dutch oven furnace is therefore of considerable importance. Their function is to provide the maximum amount of radiant heat for maintaining gasification and preventing the fire from becoming extinguished, even though the overfeed principle of continually supplying fresh fuel to the surface of the incandescent cone-shaped pile is used.

The arch location, with reference to the fuel pile, governs responsiveness to load variations. Its contour, with drop-nose at primary-furnace outlet, provides a sloping surface from which the maximum amount of heat is radiated onto the fuel pile, and at the same time forms a shield against the cooling effect of the boiler heat-absorbing surfaces.

The fuel is supplied to the furnace through openings in the arch. It has been established that a grate 9 ft square approaches the economical limit which can be supplied adequately with fuel from one feed opening. This factor, together with the known slope of the pile and the desired clearance between arch and apex of the fuel cone, makes it possible to determine the required height of arch above the grate. Individual Dutch ovens of this type are known as cells, and a steam-generating unit is provided with as many cells, all discharging their products of combustion into a common secondary furnace, as may be needed to develop required capacity. These individual cells provide better control of

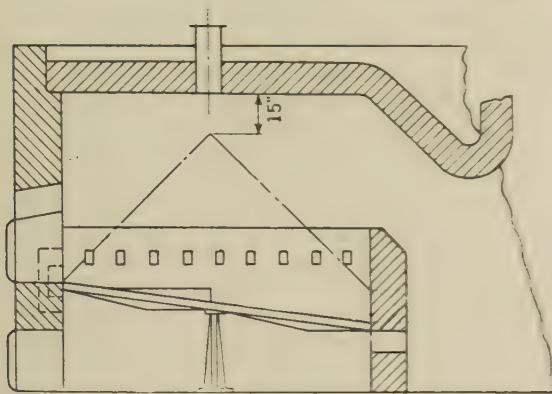


FIG. 4 SIDE ELEVATION OF A HOG-WOOD FURNACE CELL

combustion conditions and also permit periodic cleaning without shutting down the unit. A typical hog-wood cell of the type described is shown in Fig. 4.

The amount of hog fuel which may be burned in such a cell will vary from 1.5 to 3 units, depending upon wood species, whether it originated from salt- or fresh-water logs, the size of the fuel, as well as the consist of the wood mixture. These data are based upon forced-draft operation with air at room temperature. Operation can be maintained with 30 to 40 per cent excess air.

A number of installations have been made in which two or three feed openings in tandem are used to supply each cell. In these the fuel pile consists of a number of overlapping cones, corresponding to the number of feed openings. Individual cells are usually about 8 ft wide and up to 24 ft long. With this arrangement it is possible to provide greater wood-burning capacity per foot of furnace width, even though operation is not quite as satisfactory as with the smaller cells using only one feed opening.

A unit equipped with a tandem furnace, designed to burn red-wood refuse having a moisture content of 50 to 55 per cent, is shown in Fig. 5. Three cells, each having two fuel-feed openings for a grate 6 ft 8 in. wide \times 18 ft long, are used. Capacity is 100,000 lb steam per hr at 250 psi and 735 F. Burners for auxiliary oil firing are installed in the water-cooled secondary furnace, for use when no hog wood is available.

PAPER MILL BY-PRODUCT FUELS

The fuels, which constitute some of the by-products in the

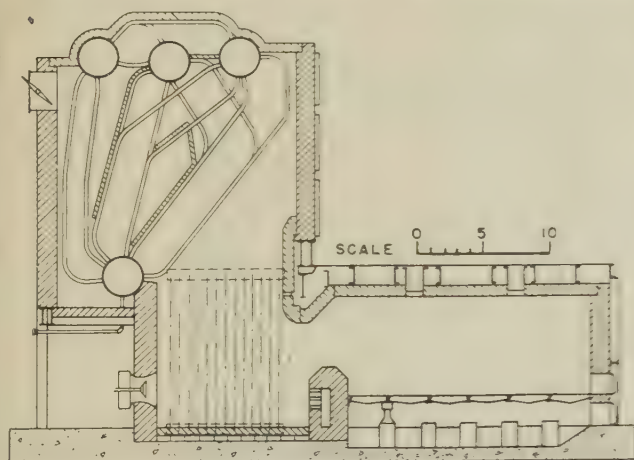


FIG. 5 STEAM-GENERATING UNIT WITH LONG DUTCH-OVEN FURNACE AND TANDEM ARRANGEMENT OF FUEL-FEED OPENINGS

manufacture of pulp, fall into two classifications: wood-room refuse in the form of wet bark, culled wood, sawdust, ground-wood screen tailings, and butt ends; and spent alkaline liquors from the chemical processes of digestion.

Wet-Wood Refuse. The most important wood-room refuse, and perhaps the most uncertain as to quantity, is wet bark. As received from the barking drums it may contain 90 per cent or more moisture. In this condition it is of no value as a fuel because its as-fired heating value is approximately 1750 Btu per lb, and for every pound of dry substance there are 4 lb of water which must be evaporated before any heat is available for steam production. If useful heat is to be realized, some preliminary dewatering must be resorted to. Pressing may be used to reduce economically the moisture content to about 65 per cent. The pressed bark can then be mixed with the other wood-room refuse and burned without further preparation. However, hogging before burning makes possible more uniform feeding and also contributes to improvements in the combustion process.

External bark driers offer a means for utilizing heat in the flue gas to effect some additional moisture removal following pressing.

The use of preheated air for combustion provides a means for securing additional drying effect in the fuel bed.

In the use of any type of drying equipment the removal of moisture from the waste fuel represents a gain in heating value, and therefore a direct saving in the use of purchased fuel or power. Likewise, an air-heater installation represents an additional reduction in purchased-fuel requirements. Comparative evaluation can be made by balancing the fuel saving against fixed charges and operating costs of the drier and air heater, respectively, to determine if the installation of one or both types of equipment can be justified economically.

In the design of furnaces for mixtures of wet bark and wood, an hourly disposal rate of 20 to 35 lb of dry solids per sq ft of grate is possible with moisture content of 65 to 70 per cent, provided auxiliary fuel is burned. For a moisture content of 55 to 60 per cent, an hourly disposal rate of 35 to 50 lb per sq ft of grate is possible without use of auxiliary fuel, and these rates may be increased somewhat through the use of preheated air or the development of improved furnace designs.

The most widely used furnaces for burning this wet-wood refuse are of the flat-grate Dutch-oven type with either single- or multiple-feed openings similar to those illustrated in Figs. 4 and 5. Auxiliary grates, oil burners, or pulverized-fuel firing are frequently used to provide additional heat for disposing of the wood waste, and to generate simultaneously, or during periods of low wood supply, such steam as may be required to meet demands on the unit.

A steam-generating unit, designed to burn a mixture of pressed bark, sawdust, shavings, and butt ends, and equipped for supplementary pulverized-coal firing is shown in Fig. 6. The wood-burning furnace consists of a two-cell Dutch oven with drop-nose arch and flat grates each 5 ft wide \times 12 ft long. The products of combustion are discharged from this primary furnace into an upper, water-cooled, secondary furnace in which pulverized coal may be burned, either simultaneously with the wood, or separately if no wood fuel is available. The wood refuse has an average moisture content of 60 per cent, and with this fuel continuous capacities of 55,000 lb of steam per hr and 15-min peak capacity of 85,000 lb per hr have been carried without difficulty. This unit has been free from slag deposits even though coal and wood are frequently burned simultaneously. Complete water-cooling of the secondary furnace side and rear walls thus provides trouble-free operation and an almost total absence of maintenance charges.

Another type of furnace which is used to burn mixtures of wet

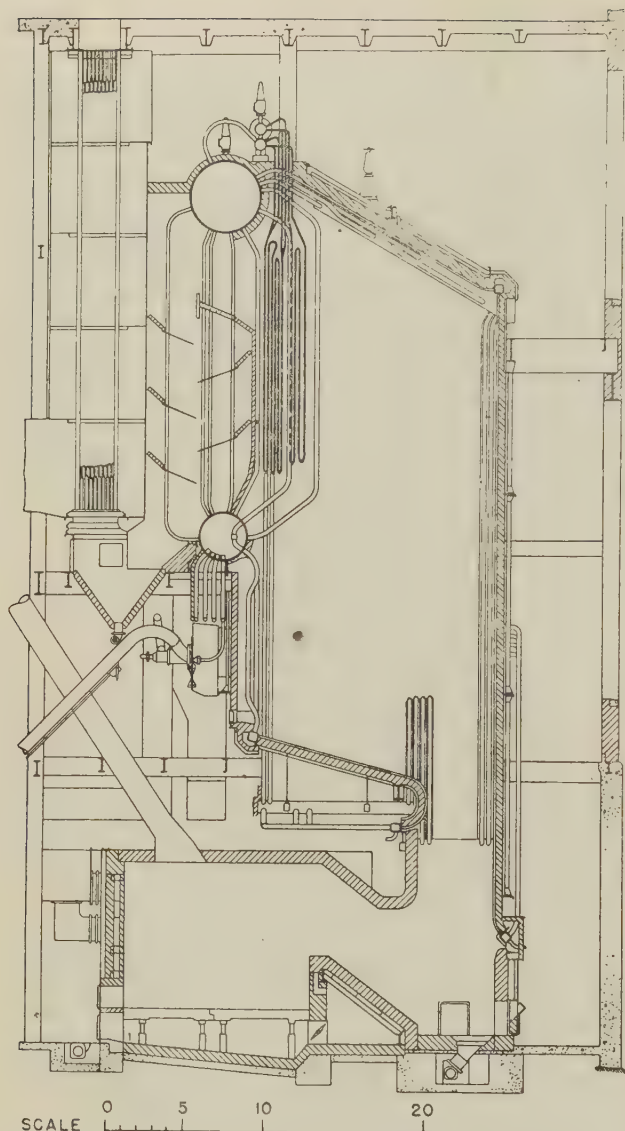


FIG. 6 STEAM-GENERATING UNIT FOR COMBINATION FIRING OF PULVERIZED COAL AND WET-WOOD REFUSE FROM A PULP-MILL WOOD ROOM

bark and wood refuse is shown in Fig. 7. With this design a sloping grate is used and the fuel enters the furnace at the front and over its full width. The fuel-supporting surfaces are divided into three sections. The upper front section forms the preliminary drying zone, and consists of a refractory hearth having a slope of approximately 50 deg. The midsection consists of stationary grates set at a slope of 45 deg and is provided with horizontal air-admitting opening. The grates of the lower section are set slightly less than 45 deg and have fuel pushers which may be operated as required. Horizontal dump grates extend from the end of the grate to the bridge wall. Progressive feeding of the fuel from entrance to dump is secured through the variation in grate slope. For large units these furnaces are sectionalized in the same manner as the flat-grate type. The Dutch oven uses a flat arch, and the opening into the secondary furnace is screened by an arrangement of cooling tubes. In the lower portion of this opening the tubes are wide spread, while in the upper portion they are finned to present a barrier to gas flow and thereby cause it to sweep over the lower end of the fuel bed. Oil burners for supplementary firing are located in the roof of the water-cooled secondary furnace.

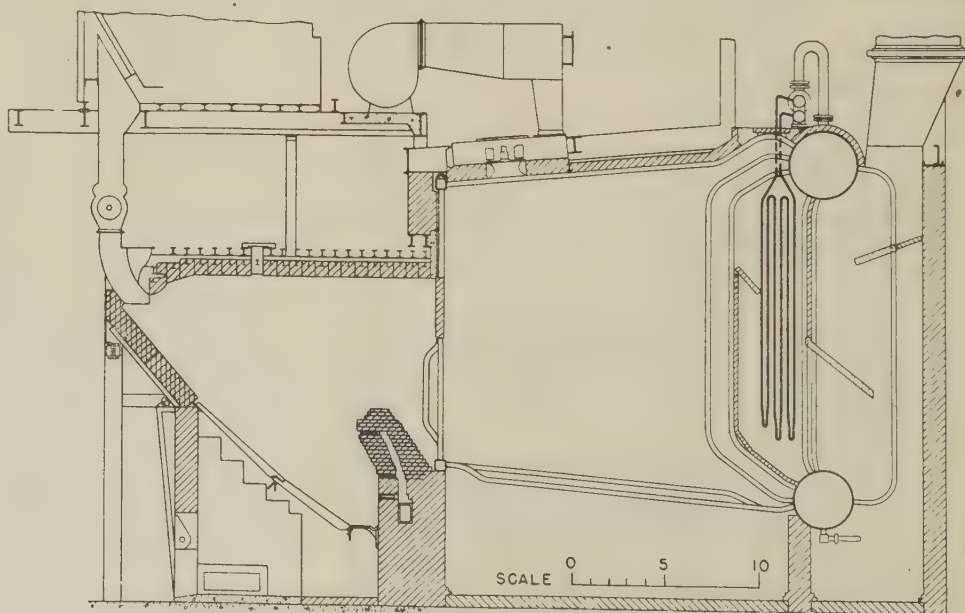
Spent Alkaline Liquors. When pulp, for the papermaker, is manufactured by the chemical processes of soda and sulphate, caustic soda is employed as the active chemical to separate the cellulose fiber, by dissolving the other wood substances. These dissolved wood substances are carbonaceous matter which is then burned in smelting furnaces to recover the chemical and use available excess heat to generate steam.

Alkaline pulp-mill operators call all chemical solutions liquors. Thus from their characteristic colors come the terms "black" liquor, the solution after the diffuser and up to the recovery unit; "green" liquor, the solution in the dissolving tank; and "white" liquor, the causticized green liquor which forms the entire cooking liquid for the digesters.

The black liquor as it is washed from the pulp and discharged to the weak-liquor tanks contains from 12 to 20 per cent dissolved solids. After passing through multiple-effect evaporators its solids content is raised to 45 to 55 per cent. Additional evaporation is next carried out in the recovery unit, after which the liquor is mixed with salt cake and then pumped to the recovery furnace where the remaining water is evaporated, the organic matter burned, and the inorganic chemicals recovered.

The amount of dry solids per ton of pulp may be as low as 2000 lb for kraft stock and as high as 3400 lb for bleached sulphate. The calorific value of black liquor varies from 5200 Btu

FIG. 7 PAPER-MILL INSTALLATION IN WHICH DUTCH-OVEN FURNACE WITH SLOPING GRATE IS APPLIED TO BOILER EQUIPPED FOR AUXILIARY OIL FIRING IN A WATER-COOLED SECONDARY FURNACE



per lb for soda liquors, up to 7000 Btu per lb for rich kraft liquors. With this range in weight and heating value, the approximate amount of steam produced will usually vary from 2400 to 3300 lb per 1000 lb of dry solids depending upon the design of the unit, and heat requirements for evaporation and chemical conversion.

The principal function of the recovery unit is to recover the chemical contained in the black liquor, and to reduce the salt cake to sodium sulphide. Carbon and organic matter are burned out of the black ash, and the resulting heat is used to smelt the chemical, and also to generate steam. The smelt runs continuously from the furnace over water-cooled smelt spouts into the main dissolving tank to form the green liquor.

The combustion process in units of this type is a critical low-temperature operation. The sodium salts in the liquor and smelt have a low temperature of vaporization and as a result some of

these are carried out of the furnace, as a gas, with the products of combustion. The temperature of the gas is decreased in passing through the boiler, and some of the soda is condensed and deposited on its relatively cold surface. The recovery of this soda is an important operation, as it must all be returned to the hearth.

In Fig. 8 is shown a completely integrated chemical-recovery unit comprising furnace, boiler, superheater, soot blowers, evaporator, salt-cake feeding and mixing equipment, fans, air- and gas-duct systems, dissolving tanks, liquor pumps, controls, and instruments.

The completely water-cooled furnace in the design illustrated extends from the hearth to the upper small boiler drum. The hearth is of chrome-refractory construction.

The lower portion of the furnace is divided into three overlapping zones between which there are no distinct lines of de-

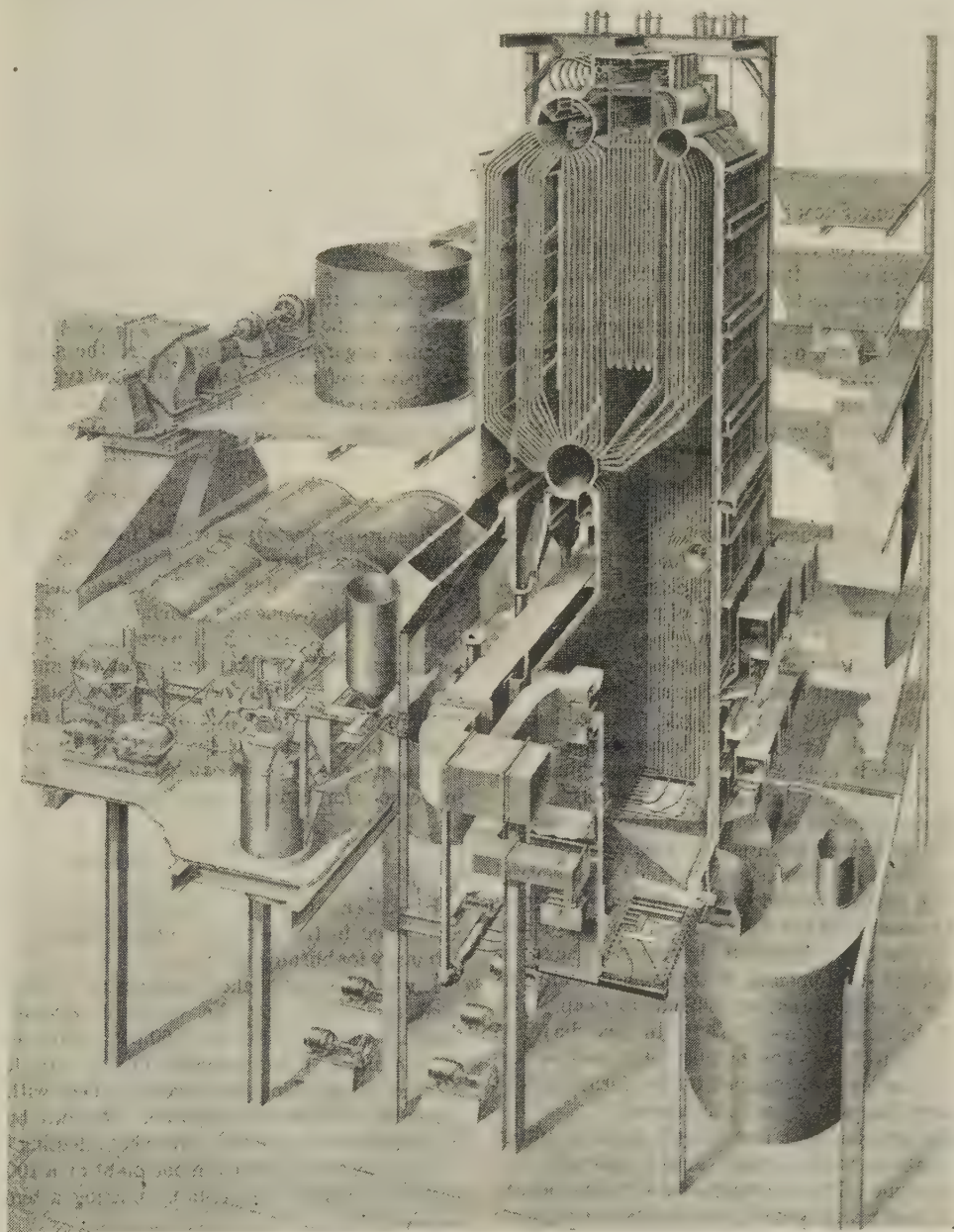


FIG. 8 CUTAWAY VIEW OF RECOVERY UNIT SHOWING DESIGN FEATURES AND STRUCTURAL ARRANGEMENTS

marcation. Immediately above the hearth a reducing atmosphere is maintained to burn the organic residue out of the black ash, and to secure maximum conversion of chemical into smelt. The intermediate zone is one in which a major portion of the moisture is evaporated from the black liquor as it emerges from the oscillating spray heads into the furnace. The heat for the evaporation process is obtained from burning some of the organic compounds out of the sprayed liquor in the upper furnace zone, and also from completing the combustion of gas leaving the reducing zone.

Gas from the last boiler pass is discharged into a cascade evaporator where the densities of the black liquor are raised from 45 to 55 per cent solids up to 65 to 70 per cent for the furnace sprays.

Although one of the principal parts of a recovery unit is the boiler section, and its general appearance follows that of standard steam-generating equipment, its distinctive function places this heat-absorbing surface in a different category. The recovery of chemicals is the primary object, while resulting steam generation is of secondary importance. For this reason, the design shown is a carefully balanced combination of chemical-process requirements and modern steam-generating practice to effect maximum chemical reduction and recovery, along with highest possible steam production.

BY-PRODUCT FUEL FROM SUGAR MANUFACTURE

Bagasse, or cane trash, is the refuse remaining after the juice is extracted from the sugar cane. It is a fibrous material, similar in analysis to wood, and contains from 40 to 60 per cent moisture. Ash content varies over a wide range depending upon the areas in which the cane is grown and on the different amounts of silt picked up by the method of harvesting employed.

Steam requirements of mills producing raw sugar are easily supplied through the bagasse they turn out. Sufficient bagasse usually remains from the crop for starting the mill at the beginning of the next season, even though the steam-generating unit is of the simplest sort.

Mills which operate white-sugar refineries are required to supplement the bagasse supply with other fuels. In these instances the steam-generating units may include economizers and/or air heaters so as to minimize the cost of purchased fuel.

Bagasse. Bagasse is burned somewhat like hog fuel in Dutch-oven-type furnaces on horseshoe-shaped hearths, or on inclined grates. The ash, which consists mostly of fine silt, is very fusible and produces a slag that is difficult to remove. These slag accumulations require periodic removal and because of this the cell type of construction is used so that the steam-generating unit may be continued on the line during cleaning periods.

The most widely used furnace is the Cook, or horseshoe type illustrated in Fig. 9. It consists essentially of a refractory Dutch oven, usually in the form of a horseshoe, provided with several rows of air-admitting tuyères located around the curved portion of the wall. The horseshoe shape was adopted because of the ease with which it is possible to distribute the fuel over the entire hearth, from a single fuel-feed opening in the arch, and also because of the absence of corners that are difficult to clean. The width of hearth may be varied from 4 ft to 5 ft 6 in., and the front-to-rear depth from 6 ft to 7 ft. The number of horseshoe furnaces used for a given boiler is determined by the furnace width available, as well as by the capacity to be developed.

Air for combustion supplied by a forced-draft fan, is admitted to the furnace through the tuyères which cause it to sweep over the surface of the fuel pile. The maximum rate at which bagasse can be burned on a hearth arrangement, as described, is approximately 350 lb of dry substance per sq ft per hr, but the most economical rate, however, is from 200 to 225 lb per hr.

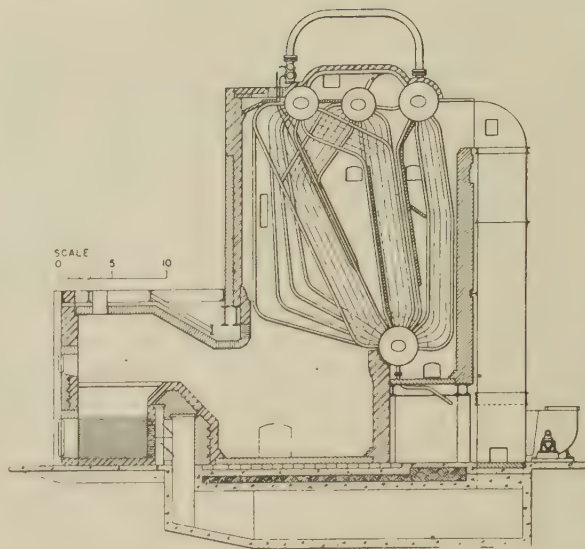


FIG. 9 STEAM-GENERATING UNIT EQUIPPED WITH HORSESHOE FURNACE FOR BAGASSE FIRING
(Tubular air heater used with this installation.)

Some installations are equipped with auxiliary oil burners in the secondary combustion chamber. These may be used to assist in starting at the beginning of the season or to provide steam during periods when bagasse is not available. It is good operating practice to avoid burning auxiliary fuel simultaneously with bagasse, in the same unit.

Some sugar-producing areas use the sloping grate in almost exclusive preference to the horseshoe type. This choice is dictated in practically all instances by the ash and silt content of the bagasse. Designs of this type are similar to those for wet wood and employ multiple-cell construction.

CONCLUSION

Present-day costs of standard fuels, such as coal, oil, and gas, have advanced to a high level, and we must now revise some of our concepts regarding so-called refuse fuels.

To-day it is frequently economical to process some of the industrial by-products which were formerly considered as necessary waste, and use them to supplement and even replace standard fuels. How well the combustion engineer is meeting this challenge is shown by the many special furnace designs which already have been developed. Others are still in various experimental stages so that we can look forward to many improvements in methods for drying, feeding, and burning by-product fuels.

Discussion

H. C. CARROLL.² To what extent can some of the by-products mentioned in the paper be used in standard furnaces designed primarily for burning solid fuel?

During the coal strike in the spring of 1947, coke breeze was used successfully by the writer in several overthrow-stoker installations furnishing up to 60 per cent of the total fuel burned on the units, when properly mixed with fast-dwindling coal storage piles. This was accomplished without reducing the capacity of the steam outputs of the units but did increase the carbon content in the ash, somewhat affecting the efficiency.

This was accomplished in one plant on a 100,000-lb unit which had six overthrow elements by putting a temporary steel-plate

² Mechanical Engineering Director, Commercial Testing and Engineering Company, Chicago, Ill. Mem. ASME.

division in the stoker hopper of each unit in the proportions of mixture desired and filling alternately, by a traveling weigh larry, the divided hoppers with coke breeze and coal. The cross action of the distributing rotors effected the proper mixing of the coal and coke of the fuel bed. A 1-month supply of coal was thus stretched to 50 days of full operation. The Btu value of coke breeze is considerably lower than coal, and usually top size is not over $\frac{5}{8}$ in.

In another plant which had parallel rail unloaders to a common apron conveyer, elevated by bracket conveyer to the bunker and then distributed to the stoker hoppers, the desired proportion of coke breeze and coal could be obtained by adjustable choke plates at the rail hopper outlets. Good mixing was thus obtained as a traveling-belt unloader on the bunker was kept in constant action, back and forth the length of the bunker. The proper proportion was determined on one furnace prior to setting up this routine for supplying the whole plant.

Dry wood waste, sawdust, sandings, etc., with the larger sizes reduced by a hog can be fed with coal on overthrow stokers if mixed properly, usually at the hopper. It can also be introduced under slight pressure direct to the furnace under control and with precautions against flashing back into the feed chute.

When industrial wastes are used as auxiliary fuels in furnaces designed for solid fuels, the economics entering into their use should be considered. Often more money is spent on the means of handling them than they will ever earn in producing steam, but credit should be given for the cost of their disposal.

In determining the value of the waste fuel, considerable care should be exercised to determine the amount available, its burning characteristics, and its heat value. We have been called on to determine this in a large range of fuels, and Table 2 of this

discussion may be of some interest in this connection. The values are mostly given on a dry basis, as the moisture may vary over a wide range, and the "as-received" value would be affected by the amount of moisture present in the material when used as a fuel.

H. W. BEECHER.³ In the northwestern portion of the United States and in British Columbia, Canada, the use of forest products has changed in the last 25 years from saw mills with a few scattering small pulp and paper plants, to a situation where practically all species of coniferous woods are now used by the pulp and paper mills in competition with the sawmills and the plywood plants.

In the pulp and paper mills, a far greater realization of cellulose content of the log for the final product is possible, and a smaller portion of the wood remains for fuel. The author refers to 50 per cent waste in manufacture of lumber. Modern pulp and paper plants now use hydraulic barkers for the removal of the bark. Larger-sized chippers that take the complete log without cutting are now prepared to take logs up to 30 in. diam. Larger logs are only sawned to sizes suitable for use in such mammoth chippers.

This modern equipment eliminates a large portion of the saw kerf formerly available for fuel. What were formerly slabs now go into pulp. Other than the bark, in the modern pulp mill from 90 to 95 per cent of the cellulose content of the original log, as received at the mill, now goes to the digesters. In the face of the increased cost of supplementary oil or coal fuel, the paper mill now faces a diminished supply of refuse from wood preparation.

With the hydraulic barker and the preponderance of bark as the log waste available for fuel, there is an increase of average moisture content in the available fuel. The amount of steam generated from the pulp-mill wood-preparation waste is but a fraction of that formerly produced. The author has clearly indicated the decrease in capacity and the decrease in efficiency resulting from high moisture content of wet wood fuel.

While we have been able to obtain capacities in excess of those cited in the paper, it has only been done with the use of specially designed furnaces in which preheated air is used and injected as overdraft air in locations that would speed up the surface combustion. Deep conical piles are undesirable. A multiplicity of small cells with individual cones in each cell have been found to be superior to those furnaces in which two or more cones are placed in series.

The conical pile is not the ideal way of burning wood fuel, but is generally adopted because of its simplicity and the ease with which fuel can be fed to the furnace. In any conical pile and for any individual rate of firing, there is a certain depth of fuel at which maximum combustion is obtainable. The center cone of the pile is always too thick and the tapering edges of the conical pile are always too thin.

If fuel could be laid down approximately to a uniform thickness and supplied with air at proper pressure, the best results would be obtained. There are several experimental plants in which bark and other mill wood refuse are being tried with spreader stokers. It is the writer's belief that the future firing of wood fuel will be in suitably designed furnaces with some form of spreader in which uniform depth of fuel can be maintained and in which the faults of the conical pile in Dutch ovens can be avoided.

The writer agrees with the author that emphasis should be placed upon the development of a furnace, boiler, air heater, and fuel drier in which the waste products will be used to increase efficiency on some of the very high-moisture fuels that are available for the pulp and paper industry.

³ Consulting Engineer, Seattle, Wash.

TABLE 2 HEAT OF COMBUSTION OF VARIOUS SUBSTANCES

Substance	Dry, Btu	Substance	Dry, Btu
Petroleum coke.....	15800	Oil (cottonseed).....	17100
#1 Gilsontite selects ^a	17699	Oil (lard).....	16740
Asphalt.....	17158	Oil (olive).....	16803
Carbon, pure.....	14544	Oil (paraffin).....	17640
Charcoal.....	14647	Oil (rape).....	17080
Carbon, crystal.....	14146	Oil (sperm).....	18000
Graphite.....	14222	Paraffin.....	18612
Pitch.....	15120	Candy.....	8096
Soot (from oil).....	11787	Butter.....	16560
Soot (from smokeless coal).....	7049	Casein.....	10548
Soot (Island Creek).....	5425	Egg white.....	10260
Soot (Red Jacket Thacker).....	10569	Egg yolk.....	14580
Soot (Crystall Block Wini- frede).....	4951	Fats (animal).....	17100
Welch anthracite (1.65 per cent ash).....	15087	Hemoglobin (blood).....	10620
Wood (beech).....	8593	Dynamite, 75 per cent.....	2322
Wood (birch).....	8588	Gunpowder.....	1314
Wood (oak).....	8316	Hydrogen gas.....	62100
Wood (pine).....	9153	Waste hemp hurds.....	7982
Wood (dry oak).....	8472	Cottonseed hulls (fusion 2342 F).....	8600
Wood (oak hog).....	8665	Cottonseed hull brans (fu- sion 2307 F).....	8675
Wood sawdust (oak).....	8493	Brown skins from peanuts.....	10431
Wood sawdust (pine).....	9347	Pecan shells.....	8893
Wood sawdust (pine).....	9676	Coffee ground (165534).....	10058
Wood sawdust (hemlock).....	7797	Pecan shells (few meats left in them).....	10144
Wood sawdust (fir).....	8249	(166838 J. W. Peterson & Co.)	
Wood sawdust (spruce).....	8449		50-50
Wood shavings.....	8248		mixture
Wood shavings (hardwood auto bodies).....	8878		coal and
Wood bark (spruce).....	8817		coke as
Wood bark (hemlock).....	8753		received
Wood bark (fir).....	9496		
Wood bark (fir).....	7999	Moisture per	
Wood bark (fir).....	4500	cent.....	21.67
Bagasse (crushed sugar cane).....	8300	Ash, per cent	13.68
Brown skins (peanuts).....	11289	Volatile, per	
Corn on the cob.....	8100	cent.....	14.01
Rags (silk).....	8391	Free carbon,	
Rags (wool).....	8876	per cent.....	50.64
Rags (linen).....	7132	Total.....	100.00
Rags (cotton).....	7165	Btu.....	9276
Cotton batting.....	7114	Sulphur.....	1.18
Corrugated fiber carton.....	5970		0.67
Newspaper.....	7883		
Wrapping paper.....	7106		
Oats.....	7998		
Wheat.....	7532		

^a Material used for cores in foundries.

C. F. HARDY.⁴ The average woodworking plant, which makes furniture, store fixtures, millwork, etc., in the Middle West, has much smaller boilers than the author mentions in his paper. Boilers may be used to provide steam for heating and process in these plants, although many also generate electric power.

The managements of the majority of these plants consider sawdust, trimmings, and hogged fuel as a waste product to be disposed of, rather than a valuable source of heat. A "unit" of 200 cu ft has a heating value of approximately 20,000,000 Btu, and 1 cu ft, which may vary in weight from 8 to 12 lb, has a heating value of 10,000 to 15,000 Btu.

In the case of wet wood from a pond or wood direct from the forest, it may be necessary to burn the sawdust and hogged fuel in the manner shown in Fig. 4 of the paper, and that is the accepted way of burning this fuel in most plants, regardless of the moisture content of the wood. It is not at all uncommon to see one of these Dutch ovens full of kiln-dried hogged fuel and shavings, and another boiler being fired by coal or oil to carry a share of the load.

On the other hand, a few plants have been equipped to fire wood fuel so that practically all of it burns in suspension, and the air is properly regulated to the amount of fuel burned. This reduces the amount of fuel burned by about 50 per cent, and thus the entire plant load may be carried without using a supplemental fuel. It is necessary only to install a storage bin (if one is not already installed) and to put in a low-cost shavings conveyer, fan, and controls, which, for the average boiler, will cost \$2500 or less. A mechanical or pneumatic spreader stoker may also be used for this purpose. Such a system has the advantage that a Dutch oven, which is used primarily as a storage space for the fuel, is not required. Generally, a furnace designed for coal or oil can be used without alteration. With industrial coal averaging from \$7 to \$10 per ton, and fuel oil costing \$15 to \$18 per equivalent ton, many plants could well afford to reappraise their opinion of so-called "wood waste" and to invest in equipment to burn it properly.

F. X. GILG.⁵ In these days of high fuel costs, the efficient utilization of by-products as fuel becomes more and more necessary. Many industries can reduce their consumption of high-cost purchased fuels by burning their by-products. At the same time, they will eliminate a troublesome disposal problem. In addition to those mentioned by the author, coffee grounds, peanut hulls, cocoanut shells, and rice hulls are a few examples of by-products with fuel value. In many instances the saving in the cost of disposing of these waste products alone will go a long way toward paying for the equipment necessary to prepare and burn them as fuel.

In the burning of coke breeze, there is considerable carry-over of partly burned cinders in the products of combustion. Unless the gas baffles are tight and carefully arranged to avoid the impingement of concentrated cinder-laden gases on the heating surfaces, severe abrasion of these surfaces may result from the sandblasting action.

In the burning of hogged fuel at high rates of combustion, particularly when the logs have been floated in salt water, severe slagging of the grates and heating surfaces may occur. To reduce trouble from this source, water-cooled grates are being used and, obviously, the walls of the secondary furnace should be water-cooled.

In the combustion of bark, particularly in the southern paper

⁴ Chief Engineer, Appalachian Coals Incorporated, Cincinnati, Ohio. Mem. ASME.

⁵ Application Engineer, The Babcock & Wilcox Company, New York, N. Y. Mem. ASME.

mills, there is a considerable quantity of sand impregnated in the bark. Slagging of the grates and boiler heating surfaces has occurred. Some of this sand is carried over with the products of combustion, causing some abrasion of the boiler tubes. Provisions must be made to settle and remove as much sand as possible from the furnace.

While there are yet many plants burning bark in Dutch ovens with flat or inclined grates, the limitations of burning in a pile and the intermittent removal of ashes makes this method of burning unsatisfactory in modern plants. A more modern design of bark-burning installation is shown in Fig. 10 of this discussion. It is a Stirling boiler with completely water-cooled furnace walls, fired by a Detroit spreader stoker with continuous ash discharge. The unit is designed to operate at 550 psi and 700 F. It will generate 85,000 lb of steam per hr when burning bark, and 100,000 lb of steam per hr when burning coal. It carries a continuous load of 95,000 lb of steam per hr on bark and has carried peak loads of 120,000 lb of steam per hr. The unit is completely automatic. The sand and ashes are discharged continuously from the front of the stoker. The cinders which are collected in the hoppers located at various points in the setting are reinjected into the furnace with the overfire-air system. This unit has been in service for 2 years. Several other units of this same type are now being constructed.

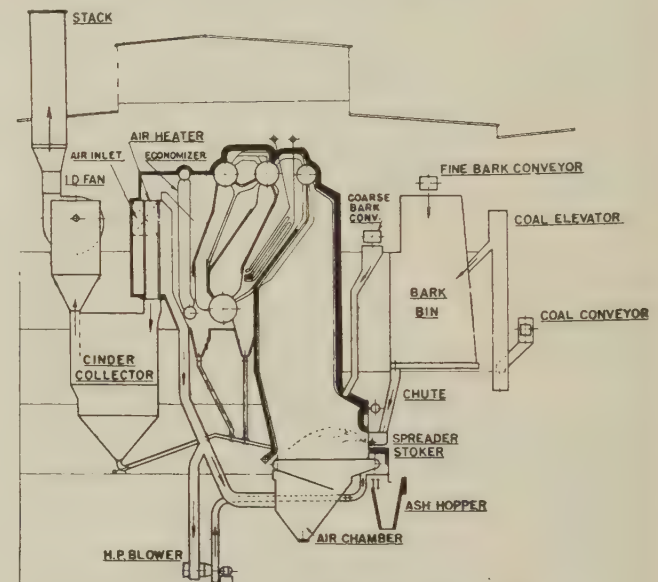


FIG. 10 BARK-BURNING UNIT WITH SPREADER STOKER

Bagasse, the sugar-cane refuse, is still being burned on piles in Dutch ovens as shown in the paper. Fig. 11 herewith shows a modern arrangement for burning bagasse at high capacities in Ward furnaces attached to the side of an integral-furnace boiler with completely water-cooled walls. It is designed for 70,000 lb of steam per hr on bagasse and 100,000 lb of steam per hr on oil or gas. The unit is designed to operate at 650 psi and 725 F. Steam loads as high as 120,000 lb per hr have been carried when burning bagasse. Carry-over of partly burned bagasse, always a nuisance in sugar mills, is almost eliminated in this arrangement. Carry-over from the Ward furnace continues to burn in suspension on its way through the water-cooled furnace of the integral-furnace boiler, a distance of about 35 ft. Some of the larger particles of course fall on the water-cooled floor which is equipped with pinhole grates supplied with air for combustion.

On a test at the design load of 70,000 lb of steam per hr, the loss in efficiency due to unburned combustible leaving the stack

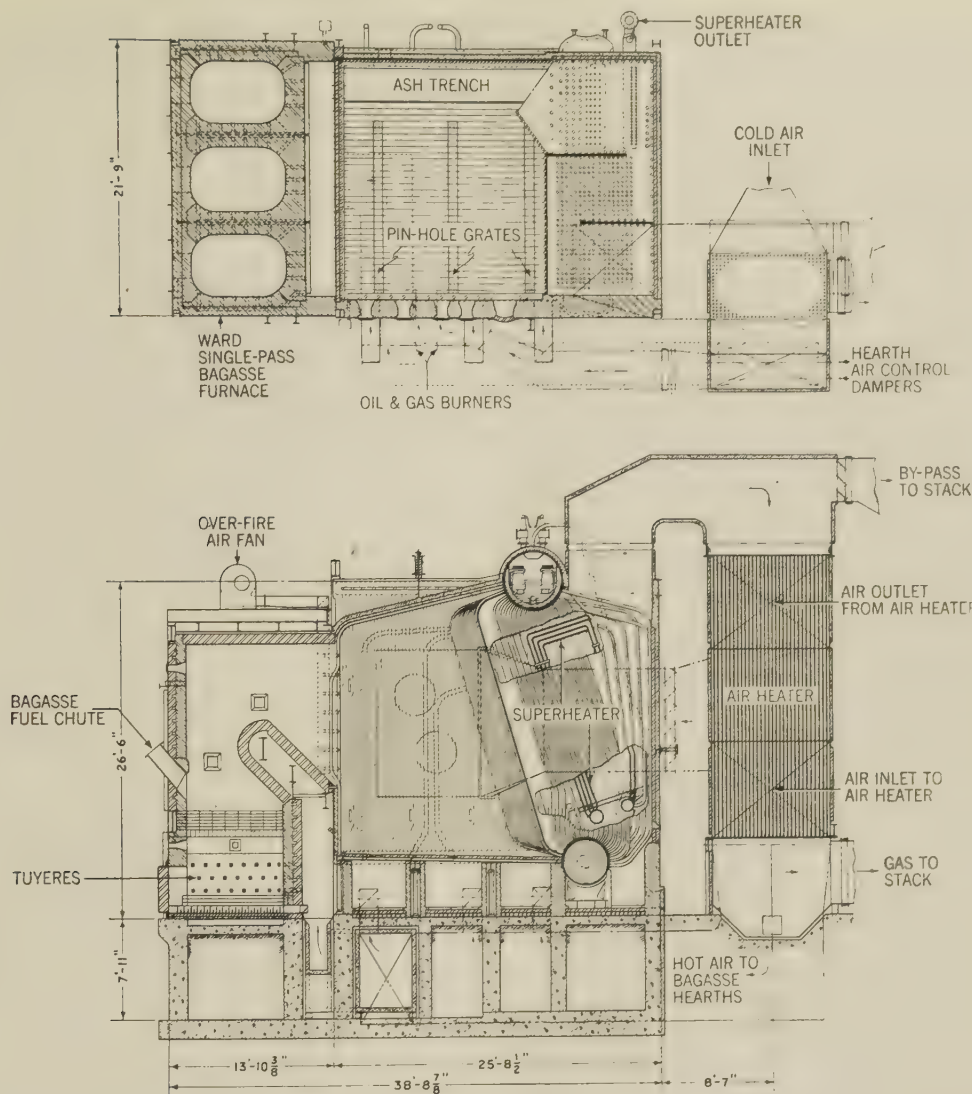


FIG. 11 ARRANGEMENT FOR BURNING BAGASSE IN WARD FURNACES ATTACHED TO INTEGRAL-FURNACE BOILER

of this unit was only 0.7 per cent. Steam pressure is maintained automatically by a Bailey combustion control, regulating fuel feed, air supply, and furnace draft. Ashes are removed from the Ward hearths once each day. Except for the manual removal of ashes, the unit is completely automatic. The high efficiency on bagasse as well as on oil is very attractive to progressive sugar-mill owners. Several other units of this same type are being constructed and in operation.

AUTHOR'S CLOSURE

The author wishes to thank the several discussers for providing additional supporting data on the use of some by-product fuels. All of these serve to emphasize the fact that many former by-products are now basic fuels because of newer furnace-design developments.

Mr. Carroll in his discussion asks: "To what extent can some of the by-products mentioned in the paper be used in standard furnaces designed primarily for burning solid fuels?" This is a difficult question to answer definitely because the so-called standard furnace is one with many variables and therefore hard to fix or define.

Where the primary solid fuel is pulverized coal it is feasible to burn acid sludge, asphaltic pitch, petroleum coke, tar, refinery gas, coke-oven gas and blast-furnace gas. The burners used must be capable of handling the by-product fuel and the capacity obtainable will depend on a number of additional factors some of which are: air and fuel capacity of the burners; available temperature, pressure, and quantity of combustion air; capacity of induced draft fan or chimney; and draft loss through the unit.

In some coal-fired spreader-stoker installations, coke breeze can be burned. In others, limited quantities of wet-wood refuse may be burned provided its moisture content is not too high and there is available preheated air for combustion.

Where chain- or traveling-grate stokers are used it is possible to substitute coke breeze for coal, provided the furnaces are not of the "archless" type.

When necessary to use a combination of solid fuels, particularly on grates of any type, it is well to avoid mixtures, because segregation will surely occur and result in unsatisfactory furnace and fuel-bed conditions. In most cases improved conditions will result from alternate, rather than simultaneous, firing of the two fuels particularly if their characteristics are quite dissimilar.

Mr. Beecher has indicated that the conical-pile method for burning wet-wood refuse while successful is nevertheless undesirable. He predicts that the future furnace designs for this fuel will employ spreader-type stokers and level fuel beds of uniform thickness. This is actually more than a prediction since considerable experimental work has already been done along these lines. Furnaces employing several variations of spreader-stoker applications, for burning wet fibrous by-products, are out of the laboratory and well along in the development stage under full-scale plant-operating conditions. Indications are that they will provide considerable increase in fuel-burning capacity along with higher efficiency and extremely long periods of service continuity. Full information on these newer developments should be available within a relatively short time.

Mr. Hardy's discussion of conditions that exist in a majority of small plants which have quantities of wood refuse available, is a timely one. In many of these even a small investment for engineering design and suitable equipment will result in appreciable savings of purchased fuel.

Difficulties which may be encountered when burning some of the by-product fuels have been pointed out by Mr. Gilg. These may be avoided when the designs used are developed through actual operating experiences.

Data furnished in connection with the bark-fired installation shown in Fig. 10 supports Mr. Beecher's predictions. Further experience, however, will surely lead to design simplification and increased capacity per unit of space occupied.

The design of bagasse furnace illustrated in Fig. 11 has been developed from the original Ward furnace of 1936.⁶ This newer so-call Ward furnace design is almost identical with one proposed and offered by the author in 1937 and later described in "Fuel Burning and Steam Generation," a book published in Canada during early 1940. The illustration of this rear-arch horseshoe-type furnace, Fig. 12, is taken from this publication.

The most recent improvements for bagasse burning now include a partially watercooled furnace construction in which Dutch-oven extensions, partition walls, arches, and horseshoe-shaped hearths are eliminated. Spreader-stoker units are used to distribute the bagasse onto either a dumping or continuous-discharge-type grate, thereby employing the combined principles of flash drying, suspension burning, and in-position burning. The fuel bed is thin, uniform, and actively burning over the entire grate. The grate surface is continuous across the length and width of the furnace, there being no partition walls, arches, or other space-consuming refractories.

A complete bagasse combustion system of this type is described by H. G. Meissner in the January, 1948, issue of *Combustion*, from which the illustration, Fig. 13, is taken. A number of furnaces of this type have now been in operation for sometime and others are under construction or on order. Carefully conducted

tests have shown increased efficiency, reduced carbon and radiation loss, and the ability to maintain easily a close control of fuel-air ratio either manually or automatically. Boiler and furnace availability are improved to the extent that these units may be kept on the line continuously throughout the grinding season.

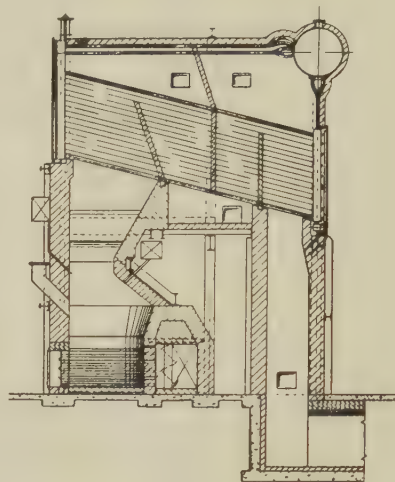


FIG. 12

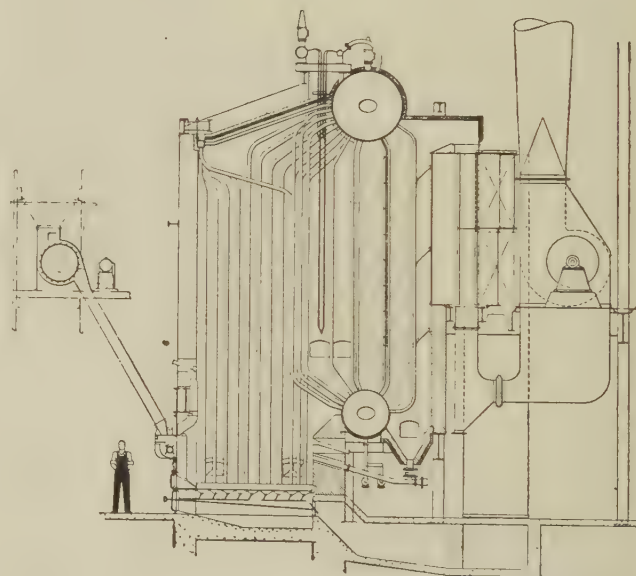


FIG. 13 C-E SPREADER-STOKER FIRING SYSTEM APPLIED TO TWO-DRUM BOILER

⁶ "Bagasse Furnace," by E. W. Kerr, Trans. ASME, 1939, vol. 61, pp. 685-691.

Proposed Recommended Practices for the Preparation of New Turbine Lubricating Systems¹

Numerous discussions between the representatives of turbine builders, operators, oil suppliers, turbine-equipment suppliers, and consulting turbine engineers in attendance at meetings of Technical Committee C, Section 1, of the American Society for Testing Materials, have indicated the desirability of pooling the available experience with respect to the preparation of new turbine lubricating systems in order that the most desirable practices might be standardized. The pooled information contained herein should be considered a "guide to desirable practices."

INTRODUCTION

IT should be emphasized that the preparation of new turbine lubricating systems on large or small units in land or marine service should be accomplished through the co-operative efforts of the turbine builder, the operator, and the oil supplier. No phase of this work should be undertaken without a thorough understanding of the possible effects on subsequent operation of the installation, nor should it be entrusted to persons lacking in experience, without adequate supervision.

While it is the primary purpose of this guide to set forth recommendations for the preparation of a system immediately prior to its initial operation, the practices followed during the manufacture and fabrication of the component parts are felt to be so closely related to the operation that a brief summary of procedures found to have given good results is set down as "manufacturers' practices."

MANUFACTURERS' PRACTICES

Piping. After fabrication in sections suitable for subsequent assembly, all piping is thoroughly cleaned by immersion in hot alkaline cleaning solutions for the removal of oil or paint. The same results may be accomplished by an alkaline steam-jet cleaning. Following this operation, the piping is thoroughly washed and pickled by immersion in hot dilute hydrochloric or sulphuric acid to which a suitable inhibitor has been added to prevent attack on the bare metal. Where bending sand has been used and may remain in inaccessible parts of the piping, the use of hydrofluoric acid for its removal is recommended. The acid-pickling operation is followed by thorough rinsing for the removal of all acid. The piping is dried, after which immediate application of a suitable rust-preventive compound is made. The use of a petroleum-base rust preventive readily soluble in oil is recommended to facilitate subsequent removal from the piping by the method recommended herein or by other methods. All open ends of cleaned and treated pipe should be sealed by a suitable means before storage or shipment. Prod-

ucts used for joint compounds or gasket materials have been known to have a deleterious effect on turbine lubricating oil. Such materials shall be free of acid, lead compounds, oil-soluble components, and other materials adversely affecting oil-service life.

Cast and Fabricated Parts Other Than Tanks or Gear Cases.

Cast and fabricated parts, such as bearing pedestals and caps, bearing brackets, valve-gear parts, etc., which are in contact with the oil, should be cleaned free from rust and scale by blasting with steel shot or steel grit, wire-brushing, or chipping. Rust-preventive coatings adequate to supply protection during shipment and storage prior to erection should then be applied.

Oil Tanks and Gear Cases. After fabrication has been completed, oil tanks should be cleaned by steel-grit or steel-shot-blasting, or by a steel brush and by a solvent to remove all grease, dirt, and other foreign matter such as joint and gasket compounds. It should then be protected either by an oil- and water-resistant paint, or by a rust-preventive coating during shipment and storage prior to assembly.

Gear casings, after fabrication has been completed, should be cleaned by steel-grit or steel-shot-blasting or by a wire brush and by a solvent to remove all dirt, grease, and other foreign matter. It should then be protected by an oil- and water-resistant paint or by a rust-preventive coating during shipment and storage prior to assembly.

CLEANING AFTER ERECTION—DIRECT-CONNECTED UNITS

General. Before any part of the cleaning job is undertaken, a thorough inspection should be made of the entire system by those charged with the responsibility for the cleaning. Its condition should be determined, and all peculiarities which may lead to difficulties, such as air pockets or areas of poor drainage, should be noted. This should be followed by an agreement on the procedure to be adopted, which should be concurred in by the builder's, operator's, and oil supplier's representatives. The initial inspection will indicate the extent of manual labor required before introduction of any oil or flushing fluid. By "manual labor" is meant the removal of thick-film (greaselike) rust preventives, brushing, blasting, and sweeping for the removal of welding shot, chips, sand, etc. and general preparation for the flushing procedure which is to follow. In this and all subsequent operations, lintless wiping cloths must be used, as accumulations of lint from this source have been known to contribute to plugging of oilways, with resultant serious damage.

Flushing. Prior to the installation and circulation of flushing oil, arrangements should be made to prevent the entrance of any contaminant carried in the oil into bearings. This can be accomplished by the installation of blind flanges at the bearing oil inlets with "jumpers" across the bearings so as to avoid long dead ends. On small installations, or in other cases where this procedure is not felt to be justified, the shells of the bearings may be removed or rotated so as to prevent flow of oil through them, but not in such a manner as to shut off the flow of high-pressure oil. Depending on the size and layout of the system, and the capacity of the pump used to circulate the flushing

¹ Report prepared by Technical Committee C, Section 1, A.S.T.M. Committee D-2; and sponsored by the Applications Subcommittee of the Petroleum Committee of the ASME Process Industries Division and presented at the Annual Meeting, Atlantic City, N. J., Dec. 1-5, 1947, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 47—A-39.

medium, it may be desirable to sectionalize the piping and flush the sections individually.

For those installations in which any rust-preventive materials are present, such materials must be removed before the final charge of lubricating oil is placed in the system. It is the responsibility of the turbine manufacturer to advise the operator or his engineers the type of rust preventive used. Rust preventives which are not readily soluble in turbine oil should be removed prior to assembly. If the rust preventive is oil-soluble and can be flushed away, it is desirable to use a flushing oil of the same or lower viscosity than the lubricating oil recommended. If this course is taken, it is considered desirable to use a flushing oil containing the same type of rust inhibitor as the final oil charge. A sufficient volume of flushing oil should be provided in the oil tanks to permit continuous circulation with the auxiliary oil pump. Means should be provided for heating the flushing oil to a temperature of 125–180 F during circulation. Heating to a temperature above the maximum to be expected in service is desirable, since it will result in expansion of the piping greater than will occur in service, with consequent more effective loosening of scale and other adhering materials.

Continuous use of a filter or centrifuge during the flushing operation is important. Where the lubricating system is not equipped with a purifier, an auxiliary filter can be installed during circulation to remove suspended contaminants which might otherwise be carried in the system. Circulation of the hot flushing oil should be continued for as long a period as necessary; this period may vary from 4 hr for small units up to 96 hr or longer for larger units. Following this flushing operation all oil is drained from the system, particular care being taken to drain all low points in the piping, coolers, and governor mechanism. Heavy solids which have been flushed into the oil reservoir are removed manually, bearing pedestals inspected and cleaned, and the governor hydraulic mechanism disassembled for the removal of any foreign material.

On small systems and those in which no rust-preventive materials have been used, the initial flushing operation can be accomplished satisfactorily with a charge of the same type oil later to be installed as lubricant. In this case the procedure will be similar to that followed with the use of flushing oil, precautions being taken to remove all loose dirt and to prevent the entrance of dirt or other foreign material into bearings and governor parts. Drainage and cleaning of the system in this case will be carried out in the manner described for flushing oil.

Displacement. In certain instances, in order to assure the complete removal of all flushing oil and any contaminants which it may carry in solution, it may be desirable to follow the flushing with a charge of "displacement" oil. The decision on necessity for use of a displacement charge should be left to the judgment of the oil supplier and turbine engineer. Factors to be considered include type and viscosity of flushing oil and general cleanliness of the system. As a displacement oil, the same type of oil as is to be used for the lubricant is installed in sufficient volume to permit circulation, and the oil, heated to 130–150 F, circulated for approximately 2 hr. Since the viscosity of this oil will be higher than that of most oils used for flushing, some particles may be suspended and flushed out which were not removed by the less viscous oil. When inspection of the centrifuge or filter at this time indicates the system to be clean, the blind flanges and jumpers which were initially installed may be removed and circulation continued for an additional 2 to 4-hr period. During this circulation, the governor mechanism should be actuated in order to assure flushing of these parts. When this operation has been completed, where practical, all accessible bearings should be removed and inspected and, if found to be in satisfactory condition, the system may be considered clean.

During drainage, attention should again be given to all low points, and the oil reservoir should be wiped out with lintless cloths, thus removing any loose solids.

In systems in which particular care has been taken during fabrication and erection, it has been found entirely satisfactory to operate the unit for a brief period up to a few weeks on the displacement charge of lubricating oil if laboratory tests indicate that the oil is in satisfactory condition, particularly in regard to viscosity and cleanliness. Where the operating requirements will permit this schedule, the difficulties frequently encountered in the operation of a new system can be worked out before the final lubricating oil is charged.

GEARED AND MARINE SETS

The initial remarks made with respect to manufacturers' practices will apply generally to direct-connected and geared sets alike; however, in view of the greater complexity required in marine systems, some variations in the procedures may be found desirable, and the following complete procedure has been found to be highly satisfactory when followed closely by a large number of shipbuilders:

Cleaning Steel Pipes, Valves, and Fittings

1 Before installation and after all fabrication has been completed, all steel pipes, valves, and fittings are to be treated as follows and in the order given:

(a) All fabricated steel pipes and flanged fittings to be blown with steam or air to remove loose scale and sand. Steel valves are thoroughly blown and then inspected to insure cleanliness. Valves should not be subjected to steps (b) through (h).

(b) All fabricated steel pipes and flanged fittings to be submerged and washed in a boiling solution of an alkaline metal cleaner such as, sodium orthosilicate. The cleansing solution should be prepared in accordance with the recommendations of the manufacturer. Pipes and fittings are to be submerged in the boiling solution for a period of approximately 3 to 4 hr. During this operation all dirt, paint, and grease are removed from the metal, thereby preparing the pipes and fittings for the acid treatment.

(c) The pipes and fittings to be rinsed in warm fresh water.

(d) The parts are then to be pickled by one of the two following methods, depending upon the time available for the pickling process:

(i) The pipes and fittings to be submerged in a solution of dilute hydrochloric acid, the composition of which shall be 1 part of concentrated hydrochloric acid and 1 part water to which a suitable inhibitor has been added to prevent attack on the bare metal. This process requires pickling for a period of 8 to 16 hr, depending upon the condition of the pipes before application of the acid treatment, or

(ii) In lieu of the dilute hydrochloric-acid solution, the parts may be submerged in an acid bath composed of one part of concentrated sulphuric acid and fifteen parts of water to which a suitable inhibitor has been added to prevent attack on the bare metal. The temperature of the solution is kept between 140 to 180 F. Parts may be pickled in this solution for about 30 to 45 min. Sulphuric-acid treatment may be preferred to the hydrochloric-acid treatment because of the fact that the former is quicker acting, and the same results may be obtained.

Either of the foregoing pickling processes removes all the scale from the metal.

Note: The experience of some engineers indicates a greater tendency toward corrosion of pipe and fitting pickled with hydrochloric-acid solution. Therefore it is recommended that wherever convenient, sulphuric acid should be used. If circumstances dictate the use of hydrochloric acid, the time of

pickling and time from the acid bath to soda neutralization (item *f*) should be kept to the minimum.

- (e) Parts should again be rinsed in warm fresh water.
- (f) Submerge parts in boiling water to which a small amount of sal soda has been added.
- (g) Rinse with cold fresh water.
- (h) Dry parts with an air blast.

Protection of Steel Pipes, Valves, and Fittings During Storage

2 After the foregoing procedure has been accomplished, the metal has a bright clean finish which is prone to rust. To prevent the rusting of the parts it is necessary to coat the steel pipes, valves, and fittings with a rust preventive for protection during storage prior to installation. The procedure is outlined as follows:

- (a) The pickled steel parts are dipped in a rust-preventive solution.

Note: Suitable oil-soluble rust preventives are preferred, since they will be removed by the flushing oil.

- (b) After the rust preventive has been applied, all open ends of the pipes, fittings, and valves are to be carefully sealed.

Bronze Valves

3 Bronze valves in lubricating-oil systems do not require the treatment as outlined in (1) and (2). Bronze valves are to be treated as follows:

- (a) The internal parts of the valves are removed and the valve bodies are thoroughly cleaned.
- (b) The valve bodies are placed in boiling water, to which a small amount of sal soda has been added, for approximately 1 hr.
- (c) Valve bodies are then rinsed in cold fresh water.

Protection of Main Reduction Gears

4 (a) The following practice has been found satisfactory for short-time storage and domestic shipment of gears and gear casings: The gears, journals, bearings, coupling, and gear casings should be cleaned with a solvent and allowed to drain, and then they should be coated with a rust-preventive compound.

If the gears are shipped assembled in the casing, no additional protection is required.

If the gears are shipped externally from the case in a crate and supported by the journals, the journals should be given special covering with greaseproof, acid-free paper and/or fabric wrapper, and additional coatings of rust-preventive compounds.

For long-time storage or foreign shipment, the gears, journals, bearings, couplings, and gear casings should be given two coatings of a rust-preventive compound. If the gear is shipped separately from the casing in its own crate, the gear teeth as well as the journal should be wrapped with greaseproof acid-free paper and/or fabric wrapper with the wrapper extending down along the rim, in order completely to protect the teeth. The wrapper should also be covered with rust-preventive coatings.

Some manufacturers prefer to coat reduction gears and the inside of the gear case with heavy grease as a protective material, in order to prevent rusting during shipment and storage prior to installation aboard ship.

Immediately prior to installation, the rust preventive is removed from the main reduction gears and the inside of the gear case and the gears are then wiped dry with clean lintless cloths.

- (b) After installation aboard ship, a small amount of rust-inhibited turbine oil is poured and brushed over the cleaned surfaces of the main gears and pinions after the gears have been checked for tooth contact. Two electric heaters are placed in the opposite corners of the sump tank to maintain a uniform temperature throughout the unit and prevent condensation of

moisture until the piping is completed and the sump tank is ready for final cleaning.

Cleaning Main Lubricating-Oil Sump Tank

5 (a) Where the oil sump is integral with the gear foundation, shall be prepared as follows prior to installation of the gear:²

- (i) The interior surface of the sump tank is washed with kerosene or other suitable cleaner to remove all foreign material and then wiped with clean lintless cloths (not waste). All surfaces are then coated with rust-inhibited turbine oil. Care must be taken that all pockets are free from water and foreign material before the system is initially charged.

- (b) After the installation of the gear and all other parts, the lubricating oil system shall be cleaned as follows:

- (i) The sump and gear casing are to be inspected and cleaned where necessary to remove foreign material. All surfaces shall be covered with rust-inhibited turbine oil.

- (ii) The two electric heaters mentioned in (4b) are removed from the lubricating sump tank.

Cleaning and Flushing Lubricating-Oil Systems

6 After the installation of all parts, the lubricating-oil system shall be cleaned and flushed as follows and in the order stated:

- (a) After the cleaning procedure has been accomplished as stated, the sump tank is ready to be charged initially with oil.

Note: Some shipyards prefer to use regular lubricating oil for the flushing operation, while others prefer the use of a special flushing oil. Some flushing oils contain kerosene and in some instances traces of rust were noted in the pipes and on the gears after using such products. Regular lubricating oil or special flushing oil, which contains a rust inhibitor, has been satisfactorily used for flushing. It is considered preferable to use flushing oil containing the same type of rust inhibitor as will be used in the final oil charge. When regular lubricating oil is used for flushing, it should be noted that all dirt and foreign material are to be removed from the oil by purifying before the oil is returned to the system.

- (b) All reduction-gear sprays and lubricating-oil leads to bearings are to be blanked off as closely as possible to the parts they serve.

- (c) To eliminate dead ends in the system, all long leads are to be connected by jumpers to the lubricating-oil drains. Jumpers should be of the same approximate diameter as the pipe being flushed.

- (d) Temporary strainers of monel metal, 40 mesh or finer, or temporary cloth bags are used on the inside of the lubricating-oil-service duplex suction and discharge strainers during the flushing period. A temporary strainer box, approximately 1/4-in. mesh, is installed around the lubricating-oil-suction bellmouth in the sump tank. Temporary strainers are installed prior to the flushing operation and are to remain in the system until and during the first dock trial. The purifier should be operated continuously during circulation of the flushing oil.

- (e) The flushing oil should be maintained at a temperature of 125–180 F for the first circulation.

The following means for heating the flushing oil may be used. All precautions should be taken to prevent localized overheating of the oil:

- (i) Low-pressure steam (3–5 psi pressure) may be led to the water side of the main lubricating-oil cooler.

- (ii) A temporary heating coil may be installed in the lubricating-oil sump tank.

² Where the sump is separate from the gear foundation it may be cleaned either before or after installation of the gear.

(iii) A temporary heater may be installed in the piping system. Coil or heater is to have no internal joints.

(f) Oil for flushing is to be pumped into the sump tanks and is to be circulated through lubricating-oil pipes without entering the bearings or reduction-gear sprayers at a temperature of 125–180 F for a period of not less than 48 hr, or until no further deposits are removed by the strainers or purifier.

(g) Experience in flushing lubricating-oil systems has shown that almost all of the larger particles of dirt and foreign material are collected in the temporary strainers during the first 2 hr of flushing. During this time it is necessary to clean the strainers at frequent intervals of approximately 15 min. The oil purifier should also be inspected frequently and cleaned if necessary.

(h) After the flushing operation is completed, the oil is pumped out of the lubricating-oil sump tank and all jumpers are removed. The discharge leads to the gear-nozzle sprays, and turbine bearings are to remain blanked off. The main lubricating-oil piping system is then tested under higher than normal working pressures to detect any leaks. The piping layout and cooler design should be checked prior to making this test to assure that no damage will be caused by the test pressures. This test is made by filling the main lubricating-oil piping systems with turbine oil and bringing the pressure up to 50 psig, or twice the working pressure, whichever is greater. Some shipyards bring the pressure up to 150 psi for a greater factor of safety. This pressure is held for a sufficient length of time to allow for examination for leaks.

Circulation of Lubricating Oil

7 (a) After completing the foregoing test all blanks are removed and the pipes adjacent to the blanks which were not flushed previously cleaned with lintless cloths (not waste) and reassembled.

(b) The lubricating-oil sump tank is inspected and cleaned if necessary. The final charge of lubricating oil is then pumped into the system.

(c) Prior to any operation of the turbines and gears, lubricating oil is circulated through the complete system at a temperature of approximately 130 F with sprays in operation and with bearing-supply needle valves wide open for approximately 48 hr or until no further solids are removed by the purifier or strainers. Strainers and purifier are cleaned periodically if necessary. Inspection plates on the reduction gears are then removed and sprayers and gear teeth inspected.

(d) After the circulation described in 7(c), a representative group of turbine bearings is examined and if found dirty, a further inspection of bearings should be made and all dirty bearings cleaned. At this time the lubricating oil in the sump tank is purified by the lubricating-oil purifier.

(e) Circulation of lubricating oil shall again be started. During this period the jacking gear shall be engaged and the main units turned over for a period of at least 2 hr. The purifier should be operated during this time.

(f) After this circulation, a representative turbine bearing and pinion bearing are to be examined. If they are not clean, inspection of all bearings should be made.

(g) After the first dock trial, transfer oil in the lubricating-oil sump tanks to the lubricating-oil settling tanks and clean the lubricating-oil sump tanks with lintless cloths (not waste).

(h) All temporary strainers are removed.

(i) After the lubricating oil is allowed to settle, it is to be run through the lubricating-oil purifier before it is returned to the main lubricating-oil sump tank.

Inspections

8 After the first dock trial, the following inspections will also be made:

- (a) Lubricating-oil sump tank.
- (b) Representative bearings, including turbine thrust bearing.
- (c) Main thrust bearing.
- (d) Gear surfaces and sprayer nozzles (through the inspection doors) and couplings and oil strainers.
- (e) Inlet end of lubricating-oil coolers.
- (f) Representative piping sections between strainers and bearing, paying particular attention to surfaces of the steel piping and internal parts of oil-actuated governor or controls.

CONCLUSION

While proper construction and preparation of the system prior to operation are essential, numerous additional factors are involved in obtaining continuous trouble-free operation of the turbine unit. These include adequate facilities for oil maintenance, proper use of such facilities, and means adopted for periodic checking of oil quality. The Committee¹ hopes to consider and possibly make recommendations in regard to these factors at some later date.

Discussion

L. BALLARD.³ The Committee is to be congratulated on its excellent approach to the establishment of a long-needed standard procedure for cleaning oiling systems of new steam turbines. The importance of an initially clean system cannot be overemphasized, particularly from the standpoint of prolonging the time until the system must again be cleaned. The oiling systems of modern steam turbines are complex, and considerable time and expense are required for cleaning and, possibly, dismantling, after the unit is in service.

Modern inhibited turbine oils will provide years of dependable service if initial contamination is eliminated, reasonable protection provided against subsequent build-up of contaminants, and the catalytic effect of metals minimized. From the latter standpoint there is convincing evidence in support of the recommendation that an oil-resistant paint or protective coating be applied to the internal surfaces of the oil reservoir either at the manufacturer's plant or prior to installing the flushing oil. Although iron is one of the least active metals in its influence on oil deterioration, nevertheless, the large area of metal surface with which the oil is in contact makes it desirable to provide a protective coating. Of even greater importance is protection of the upper parts of the reservoir against corrosion. A rust-inhibited oil will prevent effectively rusting of oiling-system parts with which it is in contact, but the upper parts of the reservoir are not so coated. As a result, they are subject to possible corrosion by water vapor and low-molecular-weight volatile acids, the latter produced as a result of oil oxidation. A suitable oil-resistant paint, a number of which are on the market, will serve the double purpose of protecting the upper parts of the reservoir against corrosion and minimizing the detrimental influence of the large area of iron surface on oil deterioration. Once the turbine is in service and the metal surfaces permeated with oil, a very rigid cleaning procedure is required to provide assurance that the paint will adhere firmly and not peel off. The proper time to apply such a coating is prior to initial operation of the unit.

C. W. BELL, SR.⁴ This paper points out a large number of

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hazards frequently overlooked in the planning for building and putting in service, equipment depending on circulating-oil systems. (The rules suggested here apply as well to heavy motor-driven equipment as to turbines.) Such a set of suggestions will assist in stabilizing what has been a very confused situation.

The paper does not emphasize the value of laboratory checking of samples during flushing and early operation of lubricating systems. This writer believes such checking valuable and would like to hear an expression from others on this point.

It has been found that by-pass filters can be notably effective if connected in a strategic manner so that heavier foreign matter is taken off naturally. This will include moisture and rust particles. The results may be indifferent if the connection is not properly located.

Every element of the lubrication system which will be either exposed to the oil stream itself or to vapors therefrom, should be blasted or pickled and properly painted, in so far as possible, with a permanent moisture- and oil-resistant paint, before being filled with oil. It is next to impossible to make paint adhere to metal surfaces after they have been oil-soaked.

Tanks and other containers, piping, etc., should be made, when it is possible, with smooth unobstructed inner surfaces. Tack-welded angles and other internal tank stiffeners interfere with satisfactory rustproofing when new and are traps for contaminants later.

The writer feels that wire brushes should only be recommended for removal of incidental rust. They will never remove mill scale nor will they produce a thoroughly satisfactory surface to which paint will adhere.

Heating of lubricating oils can be done more safely with hot water than with steam. When the more volatile solvent-type oils are used for flushing, the safe temperature limit may be critical at less than 212 F.

In general, an oil system designed to preserve the lubricating oil will be of the "slow-circuit" type, which will drop out heavy contaminants and will also have a tendency to clear itself of the usual heavier materials rather promptly.

The term "grease" as a rust preventive is questionable because many heavy greases would be bad contaminants in turbine-type oils.

LEROY F. DEMING.⁵ The following observations relative to marine installations of turbine-gear lubrication systems are based upon the writer's contact with ships built for the United States Maritime Commission during and subsequent to the war.

1 The strainers and filters through which the flushing oil is circulated serve principally to remove pipe scale, sand, and shipyard debris. A magnetic filter is usually installed to remove small particles of ferrous metals. The removal of microscopic particles of dirt, sludge, and varnish has not been observed to be a problem worthy of consideration in the flushing of new turbine and gear lubrication systems.

2 Steel pipe and plate used in the fabrication of lube-oil systems should be pickled to remove mill scale. The waiver of this requirement during the war produced results which make it appear to be questionable wisdom to depart from the requirements, even when facing a national emergency such as we faced at that time.

3 The trouble experienced with rust in some turbine-gear installations due to the condensation of moisture in the gear case was conspicuous by its absence in the C-2 ships which were fitted with a dehumidifier in the vent on the main gear case.

These units were installed on ships built by the North Carolina Shipbuilding Company at Wilmington, N. C. However, this yard was equipped with a gear shop in which the gears were assembled before being installed in the ships. The precautions exercised in this yard to insure that the gears were in good condition when installed could not be duplicated in shipyards not so equipped, and it is possible that not all of the improved operating experience with rust should be credited to the dehumidifiers.

Another feature of the dehumidifier worthy of mention is the condensation and return to the system of oil vapor which otherwise escapes into the engine room.

4 The problems arising from the use of paint to prevent rusting of parts between the time of manufacture and the time the vessel is placed in service are of interest. The vessels in the Commission's laidup or reserve fleet are prepared for layup by withdrawing the oil from the lubrication system; the system is then charged with a conditioning compound, U. S. Navy Specification 52C16a, Grade II. The conditioning compound is circulated through the system until the entire surface of the gears, the interior of the pipes, gear cases, sumps, gravity tanks, etc., are thoroughly coated. The conditioning compound which has not adhered to the surfaces mentioned is then withdrawn and re-used on other ships. No special flushing is specified for the preparation of the vessel for return to service. The conditioning compound is soluble in lubricating oil, and it is claimed it has no harmful effects. It may be applied as noted, by spray equipment or by hand brush. Since the conditioning compound has been found to be an effective rust preventive, its use in lieu of paint for the preservation of the surface of ferrous metals through the interval mentioned would seem to merit consideration.

F. S. JONES.⁶ Although it is not mentioned in the paper, it should be pointed out that the manufacturer's field engineer plays a very important part in the proper preparation of a new turbine-lubrication system prior to the installation of the operating batch of oil. He knows, through instruction from his factory, the type of material that may have been used for rustproofing of various parts of the oiling system and therefore knows what procedure should be necessary for the complete removal of such coating where necessary. Furthermore, he is informed as to whether or not any of the coatings may be of a nature that would injure turbine oil if it were used for flushing, and can decide whether or not the same turbine oil can be used for lubrication after being purified.

If various parts of the lubrication system are sectionalized for flushing, care should be taken so that no unprotected parts should be subjected to heat enough to promote condensation when they cool off again.

COMMITTEE CLOSURE

The Committee charged with the responsibility of preparing this paper greatly appreciates the comments of the various discussers relative to the need for a paper of this type. With regard to the suggestion made by several of the discussers pertaining to the application of oil and water-resistant paints to prevent rusting, the Committee has found such materials very helpful where properly selected and applied. Paints which were not suitably oil resistant or which were not applied with sufficient care have been known to contaminate the turbine lubricating oil causing operating difficulties.

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⁵ Arlington, Va.

Heat Transfer to Water Boiling Under Pressure

By E. A. FARBER¹ AND R. L. SCORAH²

The film coefficient of heat transfer from a hot metal surface to a boiling liquid was estimated from experiments with an electrically heated wire submerged in the liquid. The heat-flow rate was given by the consumption of electric power. The temperature drop through the film was taken as the difference between the surface temperature of the wire and the mean temperature of the liquid. The surface temperature of the wire was estimated by a small thermocouple. The graph of film coefficient, as a function of temperature drop through the film, is called the "boiling" curve. As the temperature drop increased, the film coefficient first rose to a maximum, then fell to a minimum, from which it rose steadily as the temperature drop continued to increase. When water was boiled at atmospheric pressure, different heated metals gave different boiling curves. Preliminary data are given for nickel, tungsten, chromel A, and chromel C. When water was boiled at different elevated pressures, the same heated metal gave different boiling curves. The data are given for nickel and chromel at 0, 25, 50, 75, and 100 psig. Throughout these experiments the general form of the boiling curve remains the same.

INTRODUCTION

THERE is an extensive literature on the subject of boiling, excellent reviews of which have been given by McAdams (1),³ King (2), Jakob (3), Drew and Mueller (4). Many investigations have been made of the boiling curve using steam-heated metal tubes submerged in various liquids boiling at atmospheric pressure. Under these conditions, the temperature drop through the boiling film has been limited to modest values, usually less than 240 F. In general, these experiments have given values along the boiling curve for various combinations of liquids and metals. These data have often been sufficient to define the maximum values of the film coefficient and frequently extend to the region of the minimum values.

Apparently it was Nukiyama (5) who first obtained values of the film coefficient in the region beyond the minimum on the boiling curve. He used an electrically heated platinum wire submerged in water boiling at atmospheric pressure. For some reason, he was unable to determine values of the film coefficient in the region between the maximum and minimum points of the boiling curve, and, consequently, his maximum and minimum values rest to some extent on speculation. Furthermore, he used no surface-standardizing procedure, as did the present

authors. Therefore his results are not comparable, so are not presented herewith for comparison.

EXPERIMENTAL ARRANGEMENTS

The fundamental characteristics of the experiment were first explored at atmospheric pressure in the open jar shown in Fig. 1. The tests reported herewith were made in the pressure apparatus shown in Fig 2, using the bus bars shown in Fig. 3. The pressure vessel consists of a 6-in. pipe-cross mounted horizontally, with glass windows fitted at two opposite flanges. One other flange is connected to a vertical 6-in. header; and the opposite flange receives the test wire and thermocouple assembly mounted on a blind flange, as shown in Fig. 3. The vertical header is provided with a water column, a safety valve, feed and drain lines, a pressure gage, a thermometer well, and an electric immersion heater.

TEST WIRES AND BUS BARS

The submerged part of the electrical circuit is shown in Fig. 3. The upper bus bar was made of 1-in.-diam cold-rolled steel shafting. One end of the bar is reduced to $\frac{5}{8}$ in. diam to pass through a $\frac{3}{4}$ -in. hole in the blind flange and threaded to receive tightening nuts and collars. The collars and the bar were elec-

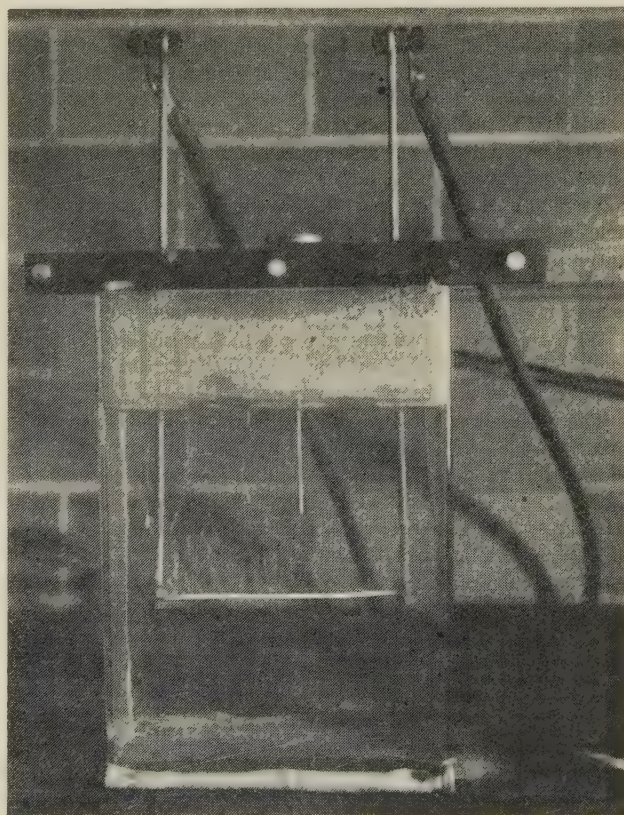


FIG. 1 ORIGINAL TEST APPARATUS

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³ Numbers in parentheses refer to the Bibliography at the end of the paper.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

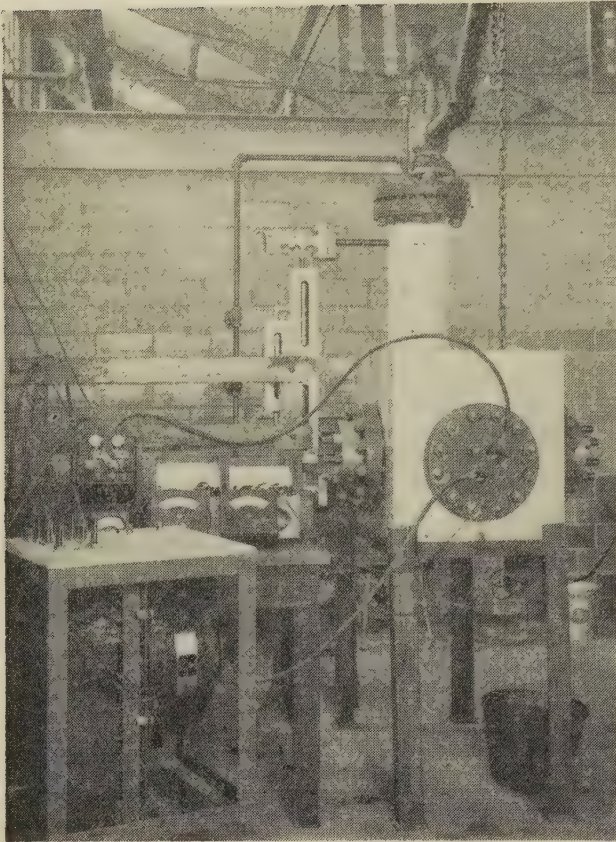


FIG. 2 PRESSURE APPARATUS USED IN TESTS REPORTED

trically insulated from the flange by gaskets made of selected India mica. A movable extension arm made of $\frac{1}{2}$ -in. \times 2-in. mild steel and provided with a locking device was fitted perpendicular to the 1-in. bar.

The lower bus bar was made of mild steel in the form of a bolt with a 2-in.-square head and a $\frac{5}{8}$ -in.-diam shank threaded for tightening nuts and collars. The assembly was made through a hole in the flange using mica gaskets. Extension bars made in various lengths of $\frac{1}{4}$ -in. \times 2-in. mild-steel stock were fastened to the square head by alloy-steel cap screws. The contact surfaces were lapped to secure good electrical connection.

With these arrangements, the bus bars were able to accommodate test wires of various lengths and to position them as desired in view of the observation windows.

The connection between the bus bars and the test wires was made by means of two compression blocks at each joint. The blocks were made of $\frac{1}{4}$ -in. \times $\frac{1}{2}$ -in. mild-steel stock each 2 in. long. The $\frac{1}{2}$ -in. faces were lapped to each other and to the bus-bar surface to insure good electrical connection. Each pair of blocks was fastened to the bus bar by two alloy-steel cap screws. The test wire was gripped for a $\frac{1}{2}$ -in. length between the two blocks. Round grooves were cold-forged in the block surface to grip the test wire without crushing. These grooves were made by pressing a pair of blocks together with a drill rod between them of slightly smaller diameter than the test wire.

In the experiments reported here, the test wires were all 0.040 in. diam and made of nickel, tungsten, chromel A, or chromel C. The length of test wire between the bus-bar connections was 6 in. in all experiments. When assembled in the pressure vessel, the bus bars held the test wires in a horizontal position at an elevation about 20 in. below the water level.

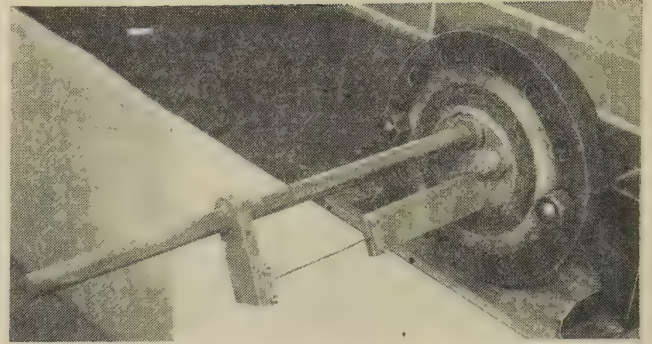


FIG. 3 BUS BARS USED IN TEST APPARATUS

The electrical resistance of the circuit was concentrated in the test wires. The end losses were small since the results obtained were independent of the test-wire length. This characteristic was demonstrated by connecting the voltmeter leads with spring clips at various positions along the length of the test wire. For a given operating condition, the voltage drop was directly proportional to the wire length. The electrical resistance of the bus bars was so low that with our instruments the same voltage drop was observed when the voltmeter leads were clipped to the bus bars or to the ends of the heated test wire. In the experiments, the voltmeter leads were connected to the bus bars.

Both direct and alternating current were used to heat the wires and no difference was detected in the results obtained. Under the conditions of our laboratory, it was more convenient to use 220-volt single-phase 60-cycle power. The supply line was connected to a 7-kva variac autotransformer which provided the quick control necessary to regulate the boiling process. The secondary of the autotransformer was connected to the primary of a 5-kva transformer having a 5 to 1 step-down voltage ratio. The 5-kva secondary was connected to the bus bars.

TEMPERATURE MEASUREMENTS

The surface temperature of the test wire was estimated in three ways: (a) by using the experimental wire as a resistance thermometer; (b) by using a small thermocouple; and (c) by an optical pyrometer at high temperatures.

When the experimental wire was used as a resistance thermometer, the measurement of the surface temperature depended upon a knowledge of the temperature field within the wire. The analysis of the temperature field required a precise knowledge of the necessary physical, electrical, and thermal properties of the wire, and precise values of the current and voltage drop. The method appeared promising, but in practice, proved to require instrumentation beyond the facilities of our laboratory. Special difficulties confronted this method as the surface temperature of the test wire approached the melting point. Evidence was obtained that the core of the wire melted first and that the melted area could spread to points on the surface without causing the wire to separate into pieces. The microstructures shown in Fig. 4 were obtained from specimens of chromel C test wires 0.100 in. diam, first in the as-received condition, and then after glowing under water for 15 min without separating into pieces. Evidently the greater part of the wire had been molten under water, and the value of such a wire as a resistance thermometer appeared very doubtful. Because of these difficulties, the use of a thermocouple and an optical pyrometer was investigated.

The thermocouples finally employed were made of chromel-alumel No. 28 gage wire. A small cylindrical thermocouple weld was used, and the couple was electrically welded to the surface of the test wire in a position normal to the test-wire

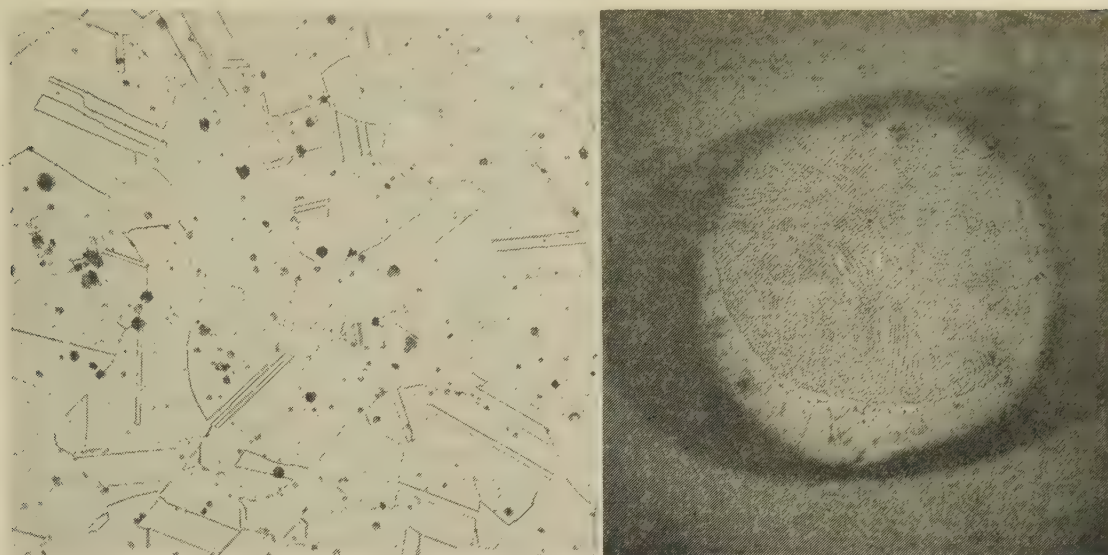
As received; $\times 250$ After glowing; $\times 18$

FIG. 4 MICROSTRUCTURES OF CHROMEL C TEST WIRES

surface. The weld was made without forming a fillet. The temperatures indicated by such thermocouples when the submerged test wire was not heated electrically agreed with the water temperatures given by a Weston thermometer. By heating the submerged test wire electrically until it was glowing, the surface temperature could be estimated by a Leeds and Northrup optical pyrometer. The results of this comparison agreed within 30 deg F at indicated temperatures of the order of 1500 F. Under these operating conditions the test wire was surrounded by a thick film of highly superheated steam. The thermocouple wires passed through this film. In so far as the eye could detect, there was no variation in color or brightness along the length of the test wire or at the thermocouple connection, and no indication of boiling was observed along the thermocouple wires. The available evidence suggests that the film of superheated steam suppressed the temperature gradient along the thermocouple wires situated within the film so that the conduction loss was small and the junction, accordingly, operated very near the surface temperature of the test wire. In the preparation of the test wires with thermocouple attached, only about one assembly in ten passed the following inspections:

1 A small cylindrical thermocouple junction welded normal to the test-wire surface without forming a fillet.

2 Agreement between the temperature indicated by the thermocouple and the temperature of the water when the submerged test wire was not heated electrically.

3 A uniform color and brightness along the test wire and at the thermocouple junction when the test wire was glowing under water.

4 Agreement within 2 per cent between the temperature indicated by the thermocouple and the temperature indicated by an optical pyrometer when the test wire was glowing under water.

Having passed these inspections, the temperature indicated by the thermocouple was taken as the surface temperature of the wire.

Test runs were made with the thermocouple attached at the top, side, and bottom of the test wire with the same results. Similar runs were made with the thermocouple attached at various locations along the length of the test wire with no difference in the results. In the experiments reported here, the thermo-

couple was attached near the mid-length of the test wire at a point about 30 deg above the horizontal. The length of submerged thermocouple wires was about 11 in.

Surface temperatures obtained in this way are certainly subject to error, but under the circumstances, they appear to be the best available estimates of the true surface temperatures.

The temperature of the water was read from a Weston thermometer and checked with the saturation temperature corresponding to the pressure.

ANALYTICAL PROCEDURE

The numerical observations consist of the length and diameter of the wire, the values of electric current, voltage drop, water temperature, and hot surface temperature.

The heat transfer is

$$q = 3.413 EI = hA \Delta T \dots \dots \dots [1]$$

and the film coefficient

$$h = \frac{q}{A \Delta T} = \frac{3.413 EI}{\pi DL (T_s - T_w)} \dots \dots \dots [2]$$

where

q = heat transfer, Btu/hr

h = film coefficient of heat transfer, Btu/(hr) (ft²) (deg F)

E = voltage drop in the test wire, volts

I = current flow in the test wire, amp

D = wire diameter, ft

L = wire length, ft

T_s = surface temperature of wire, deg F

T_w = mean water temperature, deg F

$\Delta T = T_s - T_w$ = temperature drop in boiling film, deg F

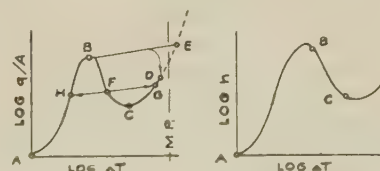


FIG. 5 TYPICAL BOILING CURVES

OPERATING CHARACTERISTICS

The general form of the experimental results is shown in Fig. 5. The boiling process was stable from *A* to *B* and from *C* to *D*, and unstable between *B* and *C*. After increasing the energy input from *A* to *B*, a further increase caused the wire temperature to rise as much as 1000 deg F, and the boiling process would attempt to come into equilibrium at some point *E* above the melting point of the experimental wire. When the energy input was quickly lowered before the wire could melt, the boiling process would reach equilibrium at some point such as *D*. The energy input could then be regulated for various conditions between *D* and *C*. Starting at *C* with slowly applied increases of energy, it was frequently possible to move the boiling process through the conditions between *C* and *B*. Starting at *F* in the unstable region, a quickly applied increase of energy would cause the boiling process to shift to some point such as *G*, and conversely, a quickly applied decrease of energy would cause the boiling process to shift to some point such as *H*. For energy inputs between *B* and *C*, there were three possible values of ΔT , the melting temperature of the wire permitting.

VISUAL OBSERVATIONS

From visual observations of the boiling process, it was possible to identify at least six different variations corresponding to different regions along the boiling curve. The transition from one type into another, though gradual, is represented by lines in Fig. 6. The different types of boiling are described as follows:

I For ΔT values from 0 up to about 4 F, steam was produced by vaporization at the liquid-vapor interface. The heat transfer from the metal surface to the liquid took place by conduction and single-phase convection which maintained an upward flow of superheated liquid.

II Ebullition began at a ΔT value of about 4 F. For values of ΔT from 4 to about 11 F, many small spheroidal bubbles would leave the metal surface, some combined to form larger bubbles, all condensed in superheated liquid before reaching the liquid-vapor interface. The boiling was nucleate in the sense that the bubbles originated at favored spots on the metal surface. Care was taken to insure the absence of subcooled liquid.

III For values of ΔT from 11 to about 65 F, nucleate boiling was observed. Larger and more numerous bubbles were generated, and they were able to transport steam to the liquid-vapor interface. Whether these bubbles increased or decreased in size could not be observed since several bubbles would almost invariably combine before reaching the interface.

IV For values of ΔT from 65 to 400 F, an unstable steam film formed around the wire, and large bubbles originated at the outer upper surface of this film. This steam film was not mechanically stable, and under the action of the circulation currents the film appeared to collapse and re-form rapidly. The presence of this steam film provided additional resistance to heat transfer, and reduced the value of the heat-transfer coefficient.

V For values of ΔT from 400 to 1000 F, the steam film around the wire was stable in the sense that it did not collapse and re-form repeatedly. The shape of the outer surface of the steam film varied continuously under the action of the circulation currents and the rapid discharge of steam bubbles.

VI For values of ΔT above 1000 F, the influence of radiation became pronounced. In this region the wire was observed to radiate visible light. The steam film was very stable mechanically, and the orderly discharge of bubbles suggested (a) that the frequency and location of bubble origination was controlled by factors operating at the outer surface of the steam film, and (b) that "favored spots" along the wire were without effect.

Fig. 7 shows in sketch form the phenomena observed in the different regions along the boiling curve. Photographic observations were tried with disappointing results.

The foregoing descriptions and sketches apply for atmospheric pressure. At elevated pressures, the corresponding values of ΔT along the boiling curve were modified as shown by the data. The appearance of the boiling process under pressure paralleled the descriptions. In general, the bubbles were smaller, the films thinner, the circulation less.

Since the same rate of heat transfer can be obtained with any one of three different values of ΔT , it was possible to operate a wire with sections at any one of three different temperatures. This type of operation was used to observe the shape and behavior of the axial end of the steam film. The middle portion of an experimental wire was heated to incandescence, and a steam film described as type VI was established. The wire on either side of this glowing section was operated at a lower value of ΔT corresponding to type III boiling. The change from film to nucleate boiling at both ends of the incandescent section was sharp. The end of the film was funnel-shaped as shown in Fig. 8. On the liquid side of the funnel, no bubbles were observed for a short distance along the wire, which suggested that the heat transferred in this section was absorbed by superheating the liquid. Only the ragged end of the funnel cone gave off bubbles, and they were discharged at a rather high velocity in the axial direction. This type of film-end boiling was a noisy process and at times caused the wire and its supports to vibrate.

EXPERIMENTAL RESULTS

Test data were taken at 0, 25, 50, 75, and 100 psig. In all tests the wires were 6 in. long, 0.040 in. diam, and were operated in a horizontal position at a point 20 in. below the water level. The results for chromel C, chromel A, and nickel are plotted in Figs. 9, 10, and 11, respectively. A comparison of the boiling curves at 0 psig for chromel C, chromel A, nickel, and tungsten is shown in Fig. 12. The test data are given in Table 1.

REPRODUCIBILITY OF EXPERIMENTAL RESULTS

When new test wires were required, they were made to glow under water for about 15 min, after which previous test results could be duplicated. One demonstration of this characteristic was the comparison of six test runs made with different wires on different days. The test data are given in Table 2, and plotted in Fig. 13. These test wires were made of chromel C, 0.040 in. diam \times 6 in. long, and were operated in a horizontal position under atmospheric pressure plus 20 in. of water.

WIRE-SURFACE-FINISH MEASUREMENTS

Preliminary studies of the wire surface have included measurements of the surface roughness of chromel C wires as-received, and after glowing under water for 15 min. The surface profiles shown in Fig. 14 were made by Mr. Paul Ogden with a Type SA2 Brush surface analyzer, and the root-mean-square roughness was measured with a type BL-105 meter. The trace was made parallel to the wire axis. The wires were 0.102 in. diam. The roughness expressed in root-mean-square microinches at 8 locations on each wire was as follows:

	Rms	Maximum	Mean	Minimum	Max/Min
As-received		54	30.5	7	7.14
Glowing 15 min.		64	43.5	23	2.78

These measurements indicate that after glowing under water for 15 min, the mean roughness of the wire surface was increased 42 per cent, and the range of roughness was decreased 61 per cent. The small increase in mean roughness was not considered significant, but the large increase in uniformity of the roughness and

TABLE 1 DATA FROM TESTS ON CHROMEL C, CHROMEL A, NICKEL, AND TUNGSTEN WIRE

(All wires 0.040 in. diam and 6 in. long. Wires held horizontal and 20 in. below water level)

E	I	Q/A	ΔT	h	E	I	Q/A	ΔT	h
April 25, 1946, chromel C, 0 psig, $T_w = 212^\circ \text{F}$.									
.029	.18	4.06	1.71	2.0	15.1	87.2	858000.	58.0	14820.
.090	.59	34.5	3.83	9.0	11.4	66.4	493000.	82.8	5930.
.43	2.14	598.	8.67	68.9	6.6	38.2	164000.	176.	932.
1.43	8.45	7860.	16.9	465.	3.94	22.6	57900.	392.	148.
4.23	24.6	67750.	26.3	2580.	3.96	22.8	58800.	591.	99.6
9.75	57.6	276000.	44.8	8020.	4.50	26.2	76800.	813.	94.3
12.95	75.2	634000.	59.2	10730.	6.40	37.2	155000.	1200.	123.
14.7	85.6	820000.	72.8	11250.	June 13, 1946, chromel C, 75 psig, $T_w = 320^\circ \text{F}$.				
16.2	94.2	992500.	99.9	9970.	.004	.026	.68	.29	.23
14.13	82.4	756000.	143.	5290.	.011	.078	.56	.64	.88
9.15	54.8	326000.	222.	1470.	.026	.154	2.61	1.37	1.90
4.33	31.6	89000.	510.	175.	.116	.671	50.7	2.22	22.8
3.99	23.2	60300.	1009.	59.8	.344	1.98	439.	3.76	118.
4.68	27.2	82900.	1420.	58.5	1.05	6.05	4130.	5.92	700.
5.95	34.6	134000.	1830.	73.2	3.35	19.4	42300.	8.98	4720.
7.90	46.0	236500.	2280.	103.5	8.55	49.6	276000.	13.3	20810.
June 12, 1946, chromel C, 25 psig, $T_w = 267^\circ \text{F}$.					12.6	67.2	551000.	16.7	33000.
.006	.032	.12	.37	.33	17.0	98.8	1093000.	22.6	48260.
.015	.086	.84	1.0	4.2	18.8	109.6	1340000.	30.1	44500.
.045	.260	7.6	1.81	22.3	15.3	89.0	886000.	42.8	20670.
.153	.885	88.2	3.94	132.	11.4	66.4	493000.	69.5	7080.
.522	3.03	1030.	7.78	598.	7.37	42.8	204000.	123.	1670.
1.40	8.10	7380.	12.3	2110.	3.89	22.6	57300.	219	262.
3.25	18.8	39800.	18.9	7013.	3.85	22.2	55600.	508.	109.5
7.08	41.0	189000.	26.9	11230.	4.50	26.2	76800.	691.	111.
9.67	59.6	376000.	33.4	16120.	5.96	34.6	134000.	889.	151.
13.50	78.4	690000.	42.9	18090.	June 15, 1946, chromel C, 100 psig, $T_w = 337^\circ \text{F}$.				
15.60	90.6	920000.	51.1	17250.	.006	.043	.17	.34	.49
16.75	97.4	1063000.	61.7	6320.	.018	.106	1.24	.70	1.79
13.60	79.0	700000.	111.	455.	.058	.337	12.7	1.32	9.7
5.77	33.4	125400.	276.	81.1	.218	1.27	180.	2.21	81.8
3.80	21.2	52500.	643.	73.5	.858	4.97	2780.	3.46	805.
4.45	25.6	74200.	1007.	81.9	3.93	22.7	58000.	5.68	10200.
5.57	32.0	116200.	1420.	109.3	10.0	58.2	379000.	7.30	51830.
7.28	42.2	200000.	1830.	109.3	14.7	85.6	819000.	9.20	89160.
June 13, 1946, chromel C, 50 psig, $T_w = 297^\circ \text{F}$.					20.5	119.0	1590000.	12.8	124200.
.0019	.114	.14	.90	1.57	20.0	116.4	1564000.	16.1	94250.
.053	.310	10.7	1.72	6.23	18.1	105.0	1240000.	19.9	62100.
.182	1.06	125.6	3.21	39.2	14.5	84.6	798000.	24.7	32500.
.542	3.14	1109.	5.68	195.	8.9	53.4	309000.	50.9	6050.
1.67	9.70	10540.	9.72	1086.	4.25	24.8	68600.	87.1	789.
4.78	27.7	86200.	15.6	5520.	2.75	15.8	28300.	178.	159.
7.89	45.8	235000.	19.8	11930.	2.89	16.8	31600.	251.	126.
12.2	71.2	566000.	25.7	22080.	4.42	25.6	73700.	432.	171.
14.9	86.4	839000.	32.1	26170.	May 6, 1946, chromel A, 0 psig, $T_w = 211^\circ \text{F}$.				
15.5	89.8	906000.	39.5	22910.	.003	.017	.03	.19	.17
					.005	.022	.07	.31	.23
					.012	.057	.44	.58	.77
					.076	.126	6.23	1.00	6.23
					.061	.296	11.75	1.70	6.9

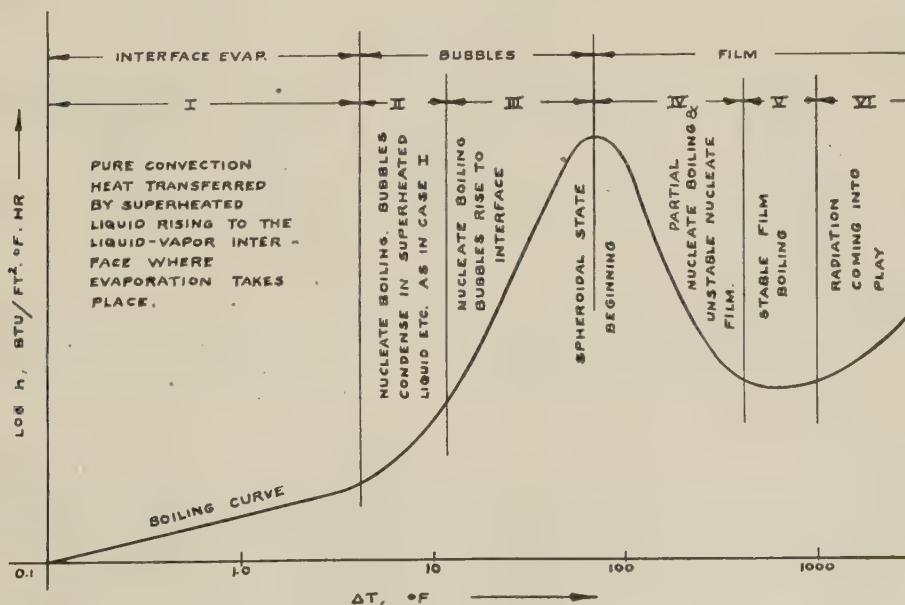


FIG. 6 PHYSICAL INTERPRETATION OF BOILING CURVE

TABLE 1 (Continued)

E	I	Q/A	ΔT	h
5.65	27.6	101600.	9.02	11250.
4.01	19.46	50900.	13.5	3760.
2.54	12.36	20420.	20.8	981.
2.74	13.36	23810.	39.8	600.
5.17	25.2	84900.	14.2	598.
8.25	40.2	216000.	239.	906.

May 10, 1946, nickel, 0 psig, $T_w = 212^\circ \text{F}$.

.00188.	.1196	.15	.35	.42
.00388	.2465	.62	.62	1.00
.00872	.558	3.17	1.09	2.92
.0384	2.44	61.1	2.37	25.8
.0848	5.39	297.6	3.22	90.3
.178	11.38	1320.	4.63	286.
.495	31.35	10100.	6.22	1630.
.924.	58.4	35120.	7.58	4630.
1.56	99.4	101000.	8.99	11210.
2.51	159.4	260500.	11.8	22130.
2.90	185.0	349000.	14.3	26500.
3.03	193.0	380600.	18.3	20810.
1.89	120.8	148700.	30.3	4920.
.83	52.6	28400.	65.1	436.
.67	42.4	18500.	222.	83.
.87	55.6	31440.	438.	72.
1.13	72.0	52900.	698.	76.
2.30	146.0	218500.	1750.	125.
3.03	193.0	380500.	2080.	183.

Juen 29, 1946, nickel, 25 psig, $T_w = 267^\circ \text{F}$.

.0014	.090	.08	.28	.30
.0065	.415	1.76	.82	2.14
.0298	1.848	35.9	1.78	19.6
.123	7.86	629.4	3.08	205.
.352	21.62	4954.	4.61	1030.
.985	62.8	40270.	5.99	6720.
2.43	154.8	244900.	8.73	28150.
2.98	190.0	368600.	11.1	33210.
3.09	197.0	1038000.	13.2	30150.
2.17	138.4	195500.	20.7	9470.
1.06	67.6	46650.	39.5	1120.
.699	44.6	20300.	123.	165.
.915	58.2	34670.	382.	91.
1.15	73.0	54650.	619.	88.
2.13	135.8	188300.	1420.	133.
2.73	238.0	423000.	2090.	277.

July 1, 1946, nickel, 50 psig, $T_w = 297^\circ \text{F}$.

.00188	.119	1.46	.32	.46
.00519	.369	1.25	.67	2.07
.0206	1.308	17.54	1.33	13.2
.1048	6.65	453.7	2.22	205.
.421	26.82	7350.	2.98	2480.
1.31	83.6	71300.	3.99	17930.
2.43	154.6	244500.	5.48	44720.
3.34	212.0	461000.	7.21	64340.

TABLE 1 (Continued)

F	I	Q/A	ΔT	h
3.61	230.0	540500.	10.6	51110.
1.63	103.6	109900.	20.3	5420.
.882	56.2	32270.	52.1	620.
.810	51.8	27310.	203.	135.
1.09	69.2	49100.	462.	106.
1.53	97.6	97210.	821.	119.
3.45	220.0	494100.	1730.	287.

July 2, 1946, nickel, 75 psig, $T_w = 320^\circ \text{F}$.

.00123	.0787	.06	.22	.29
.00427	.2718	.76	.52	1.46
.00851	5.42	30.1	1.10	27.3
.0354	22.61	522.	1.68	311.
.568	36.2	13400.	2.25	5950.
1.56	99.2	101000.	3.14	32110.
3.02	192.4	378500.	4.63	80170.
3.72	236.0	571000.	5.58	102300.
3.96	252.0	650000.	6.92	94250.
4.16	264.0	715000.	11.9	60320.
2.03	129.0	170800.	17.3	9870.
1.20	76.6	59000.	28.5	2105.
.89	56.4	32700.	122.	267.
1.11	70.8	51200.	359.	143.
1.94	124.0	156900.	830.	189.
3.13	199.6	407000.	1480.	276.

July 3, 1946, nickel, 100 psig, $T_w = 337^\circ \text{F}$.

.00278	.177	.32	.35	.91
.0123	.783	6.27	.61	10.3
.0636	4.04	167.3	.92	182.
.288	18.3	3439.	1.32	2610.
.965	61.6	38700.	1.78	21800.
2.18	131.8	187000.	2.16	86400.
3.65	232.0	552000.	3.07	181200.
4.03	256.0	672000.	4.31	156000.
2.68	170.4	298000.	7.80	38200.
1.37	87.4	78000.	19.2	4080.
.910	57.8	34250.	37.9	906.
.911	58.0	34450.	178.	193.
1.09	69.2	49200.	293.	168.
1.64	104.4	111500.	618.	181.
3.22	205.0	431000.	1390.	310.

May 12, 1946, tungsten, 0 psig, $T_w = 212^\circ \text{F}$.

.0021	.0326	.05	.25	.18
.0041	.753	.20	.48	.51
.0142	.214	1.98	1.11	1.79
.0322	.488	10.23	1.98	5.18
.0762	1.152	57.1	3.16	18.1
.20	3.044	396.	5.61	70.9
.45	6.81	1995.	8.21	243.
1.00	20.0	13020.	13.0	832.
2.00	30.0	39060.	14.0	2315.
2.50	40.0	65100.	16.0	3370.

TABLE 1 (Continued)

E	I	Q/A	ΔT	h
3.25	50.0	105800.	17.0	5170.
4.10	60.0	160100.	18.0	8470.
4.70	70.0	214200.	19.0	9350.
5.25	80.0	273400.	20.0	11340.
6.00	92.0	359300.	21.0	14200.

TABLE 1 (Continued)

E	I	Q/A	ΔT	h
1.58	.772	79.4	2.75	29.
.379	1.35	158.	4.20	109.
.938	4.48	2730.	6.10	2100.
2.42	11.8	18600.	8.90	8050.
5.97	29.0	112600.	14.0	20000.
9.87	47.6	306000.	15.2	25500.
12.12	59.2	466000.	18.3	24500.
13.89	47.6	610000.	25.0	22100.
13.92	66.2	618000.	28.0	9100.
10.15	49.4	327000.	36.0	2600.
6.40	31.0	129000.	50.	537.
3.53	13.0	36000.	67.	165.
2.39	11.6	18100.	110.	72.
2.3	11.3	24700.	260.	70.
2.79	12.6	37600.	352.	72.
3.43	16.8	89000.	520.	98.
5.30	25.8	286000.	910.	195.
9.50	46.2	840000.	1460.	491.
16.3	79.2		1710.	

June 25, 1946, chromel A, 25 psig, $T_w = 267^\circ F$.

E	I	Q/A	ΔT	h
.0046	.022	.07	.23	.29
.0093	.045	.27	.41	.66
.020	.099	1.29	.67	1.95
.044	.184	5.27	1.06	4.97
.109	.535	38.	1.91	20.
.254	1.243	206.	2.83	73.
.661	3.24	1390.	4.30	325.
1.563	7.65	7790.	6.10	1280.
2.91	14.2	26900.	7.90	3410.
5.47	26.6	94800.	10.0	9500.
8.50	41.4	229000.	12.5	18310.
10.00	48.8	318000.	13.2	24060.
11.62	58.0	438000.	14.1	31100.
13.3	64.3	561000.	15.6	35970.
14.45	70.4	682000.	17.2	38670.
13.55	66.0	582000.	21.5	27100.
12.05	53.6	461000.	25.1	18320.
6.89	33.6	151000.	37.5	4010.
4.05	13.8	52200.	57.2	912.
2.84	13.8	25800.	97.5	263.
2.83	14.0	26200.	132.	145.
3.44	16.3	37600.	323.	116.
5.11	25.0	83200.	561.	143.
7.77	39.8	191000.	905.	212.
16.6	80.4	867000.	1480.	586.

June 26, 1946, chromel A, 50 psig, $T_w = 297^\circ F$.

E	I	Q/A	ΔT	h
.0032	.016	.03	.17	.19
.0061	.030	.12	.30	.40
.017	.063	.36	.53	1.72
.055	.275	9.85	1.10	8.9
.160	.324	34.9	1.15	16.2
.284	1.387	255.	2.15	65.5

TABLE 1 (Continued)

E	I	Q/A	ΔT	h
1.054	5.16	35400.	6.1	5820.
7.32	35.6	170000.	8.3	20500.
10.4	50.4	341000.	9.8	31800.
13.4	65.2	568000.	11.8	48100.
15.5	75.0	742000.	13.9	53400.
15.8	77.0	792000.	15.6	50800.
9.38	45.6	278000.	29.5	9470.
5.06	24.6	81000.	47.3	1690.
3.40	16.6	36800.	96.2	378.
3.42	16.8	37400.	168.	221.
3.67	18.0	43000.	229.	187.
4.23	20.6	56700.	306.	185.
4.90	23.2	75900.	402.	186.
6.62	32.2	139000.	590.	236.
9.37	45.6	276000.	812.	343.
15.9	77.8	805000.	1192.	673.

June 27, 1946, chromel A, 75 psig, $T_w = 320^\circ F$.

E	I	Q/A	ΔT	h
.0042	.020	.05	.20	.28
.0112	.055	.40	.39	1.33
.0321	.153	3.3	.59	4.31
.0943	.445	22.9	1.12	20.5
.2265	1.103	163.	1.83	89.2
.993	2.41	1550.	2.31	673.
1.46	6.11	5800.	3.03	1890.
2.692	13.13	23100.	3.87	5990.
5.32	23.4	107500.	5.1	21100.
11.55	56.4	426000.	6.3	67200.
15.55	75.2	767000.	7.6	101200.
17.67	86.2	990000.	9.9	111300.
18.4	89.5	1077000.	10.1	109000.
17.65	83.2	990000.	17.6	93200.
9.22	45.0	237000.	17.6	15300.
6.41	31.2	130300.	21.3	5930.
4.72	23.0	70800.	23.2	2510.
4.07	19.5	52500.	36.5	1430.
2.90	14.2	26800.	39.2	661.
4.15	20.2	54500.	151.	361.
5.27	25.8	63500.	296.	299.
7.36	35.8	172000.	503.	342.
11.90	58.2	451000.	760.	596.

June 28, 1946, chromel A, 106 psig, $T_w = 370^\circ F$.

E	I	Q/A	ΔT	h
.0087	.043	.24	.31	.79
.0266	.130	2.25	.55	4.1
.0795	.390	20.2	.88	23.
.226	1.104	162.5	1.32	123.
.653	3.20	1361.	1.61	848.
1.938	9.45	11920.	2.02	5920.
4.70	22.8	69750.	2.41	28900.
9.15	44.6	266000.	2.97	89600.
10.80	52.6	370000.	3.21	115200.
14.60	71.2	676000.	3.92	170300.
16.20	78.8	831000.	4.32	192200.
16.22	79.2	836000.	5.71	146700.
13.58	66.2	585000.	6.29	93500.

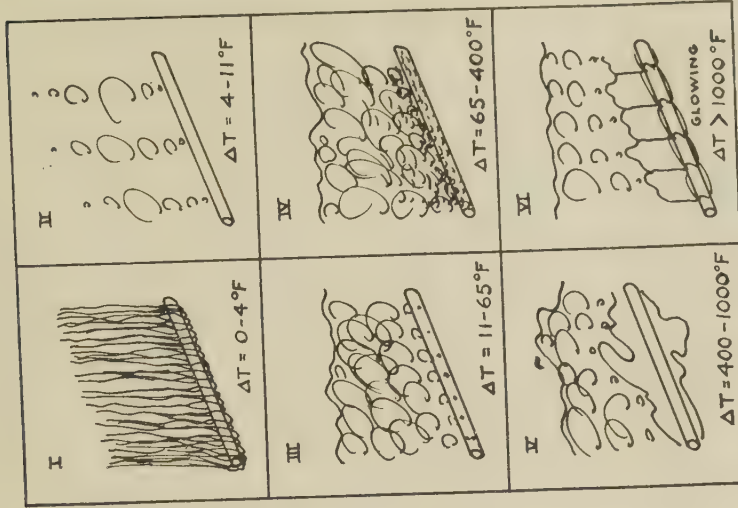


FIG. 7 TYPICAL BOILING FOR THE DIFFERENT SECTIONS

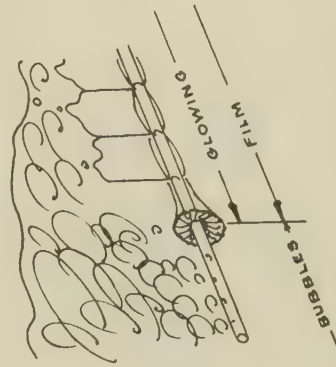


FIG. 8 FUNNEL SHAPE AT END OF FILM

TABLE 2 COMPARATIVE DATA FROM SIX TEST RUNS ON DIFFERENT WIRES ON DIFFERENT DAYS
(Six chromel C wires with pressure 0 psig. All wires 0.040 in. diam and 6 in. long. Wires held horizontal and 20 in. below water level)

E	I	Q/A	ΔT	h	E	I	Q/A	ΔT	h
Wire No. 1, April 10, 1946, Tw = 212° F.					Wire No. 3, April 18, 1946 - Tw = 212° F.				
.00182	.0106	.01	.12	.10	.00326	.0189	.04	.22	.18
.00228	.0132	.02	.13	.15	.00747	.0432	.21	.51	.41
.00557	.0322	.12	.39	.30	.024	.138	2.15	1.39	1.55
.0117	.0678	.52	.76	.68	.103	.598	40.2	4.10	10.2
.0496	.287	9.27	2.26	4.1	.465	2.72	832	9.30	88.5
.160	.926	90.7	5.05	19.2	1.77	10.3	11890	18.8	632
.712	4.12	1910	11.1	172	5.41	31.2	110000	28.9	3810
2.29	13.3	19980	20.1	993	7.52	43.8	214800	34.1	6300
4.97	28.8	93500	29.1	3210	11.6	67.6	512500	51.0	10050
6.05	35.2	138800	34.0	4080	14.1	82.2	755400	67.5	11200
9.27	53.6	324000	40.5	8000	16.1	93.2	977000	81.0	12060
10.10	59.0	388800	48.0	8100	15.7	84.4	865000	115	7520
11.25	65.6	481000	54.0	8900	7.13	41.4	190200	24.1	790
13.25	76.8	665000	60.1	11050	4.70	27.4	84000	410	205
15.60	90.4	915000	84	10900	3.60	21.0	49300	783	63
14.60	84.4	801000	109	7350	4.10	23.8	63800	1160	55
14.40	83.6	783000	151	5190	4.88	28.2	89600	1690	53
11.10	64.2	465000	193	2410	7.56	44.0	216500	2165	100
5.43	31.4	111200	372	299	Wire No. 4., April 19, 1946, Tw = 212° F.				
3.79	23.8	58800	588	100	.00171	.0099	.01	.10	.11
3.78	21.8	53600	1050	51	.00219	.0127	.02	.15	.12
3.84	19.8	109000	1880	58	.00277	.0162	.03	.19	.15
6.67	37.4	164000	2030	82	.00831	.0482	.26	.58	.45
					.0518	.300	10.1	2.30	4.3
					.216	1.26	177	5.92	30
Wire No 2., April 11, 1946, TW = 212° F.					3.25	18.9	40000	24.7	1620
.00261	.0150	.02	.18	.14	3.84	22.4	55950	25.3	2210
.00675	.0392	.17	.49	.35	6.05	35.4	139300	30.9	4510
.0173	.0998	1.15	1.10	1.05	7.09	47.8	256000	39.2	6530
.105	.438	15.6	2.79	5.61	11.35	65.6	485000	51.8	9370
.178	1.02	113	5.59	21.1	13.30	77.0	666000	60.0	11100
.880	5.08	2910	13.1	222	15.80	91.6	942000	77.8	12100
2.89	16.8	31600	22.9	1380	16.20	94.0	994000	109	9120
6.12	35.6	141900	31.8	4460	14.65	85.2	813500	133	6110
6.68	54.0	235000	38.2	6150	12.80	74.2	617500	172	3590
12.0	69.6	546000	53.5	10200	5.85	40.2	153100	287	533
13.0	75.6	642500	64.2	10050	3.94	23.0	59100	519	114
15.3	88.4	877500	88.0	9970	4.00	23.2	60500	605	100
14.8	86.2	834500	126	6620	3.65	21.2	50400	710	71
13.8	79.8	715500	159	4500	3.99	23.0	59800	1220	49
12.1	70.4	555000	178	3120	6.45	37.6	158000	2080	76
5.85	33.9	129000	322	401	8.10	47.0	248000	2215	112
3.67	21.4	51200	610	84	Wire No. 5., April 25, 1946, data given in Table 1.				
3.94	22.8	58500	1220	48					
7.22	42.0	197500	2100	94					

TABLE 2 (Continued)

E	I	Q/A	ΔT	h
Wire No 6., May 12, 1946, TW = 212° F.				
.00208	.0121	.02	.14	.12
.00276	.0161	.03	.20	.14
.00485	.0280	.09	.31	.28
.0144	.0834	.78	.91	.86
.0398	.230	5.96	1.93	3.09
.0737	.427	20.5	3.05	6.72
.143	.828	77.1	4.56	16.9
.324	1.88	398	7.65	52
1.255	7.26	5945	15.2	391
3.50	20.2	46160	22.2	2080
5.79	33.4	126000	30.6	4010
7.55	43.8	216000	35.4	6100
10.7	61.8	430000	46.2	9310
13.6	78.4	694000	62.5	11100
15.7	91.2	934500	89	10500
13.6	79.0	701000	169	4150
6.20	35.8	144600	295	490
3.56	20.8	48300	895	54
4.58	26.8	80050	1510	53
6.86	39.8	178000	2170	82

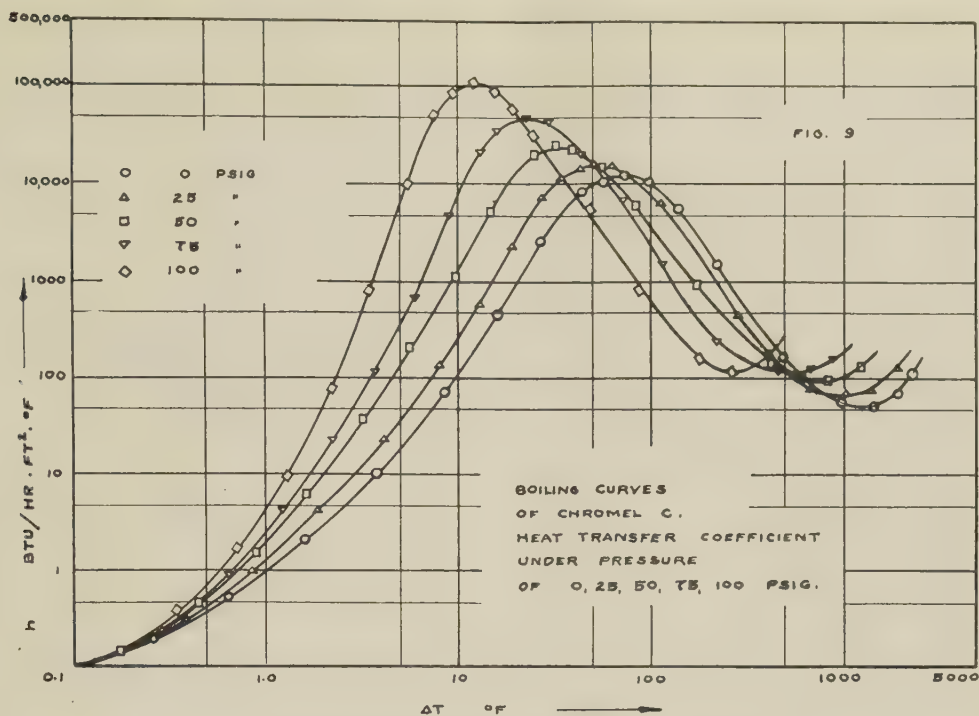


FIG. 9 BOILING CURVES OF CHROMEL C TEST WIRE

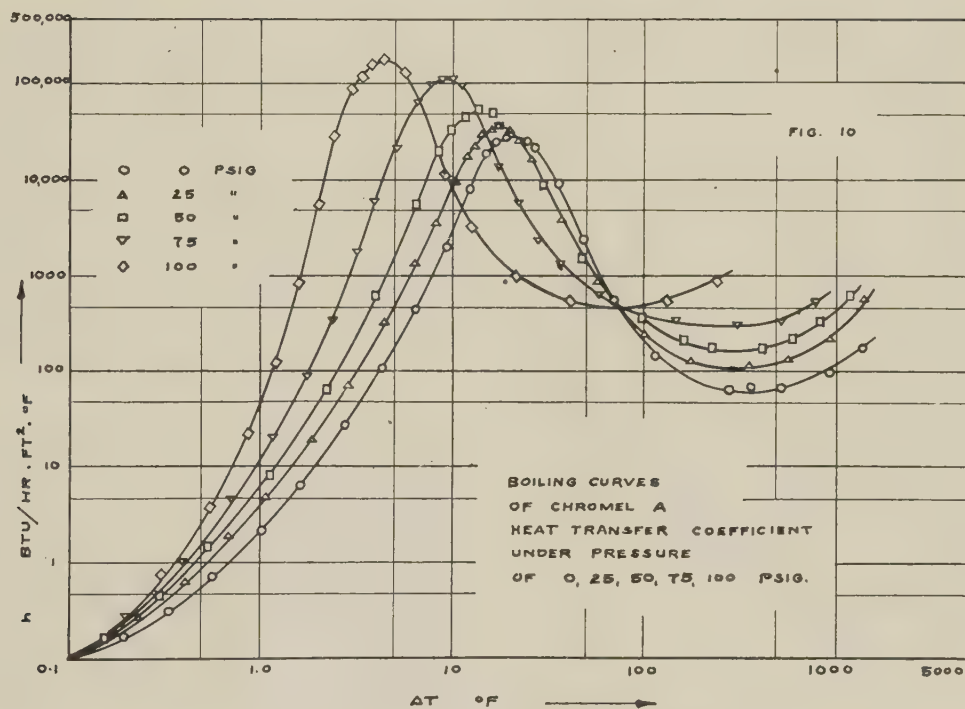


FIG. 10 BOILING CURVES OF CHROMEL A TEST WIRE

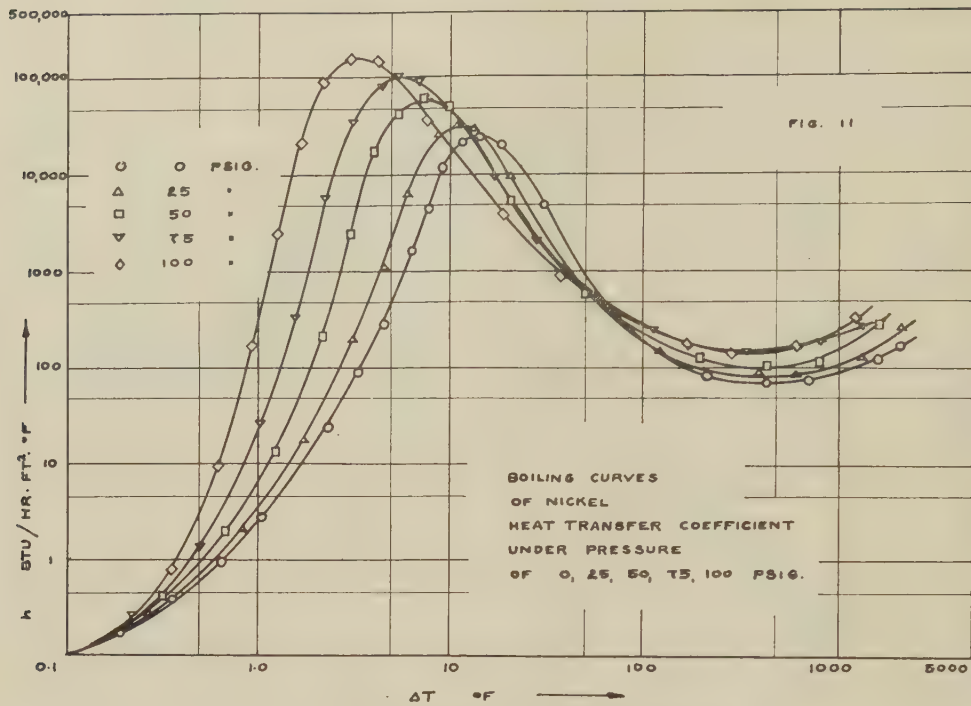


FIG. 11 BOILING CURVES OF NICKEL TEST WIRE

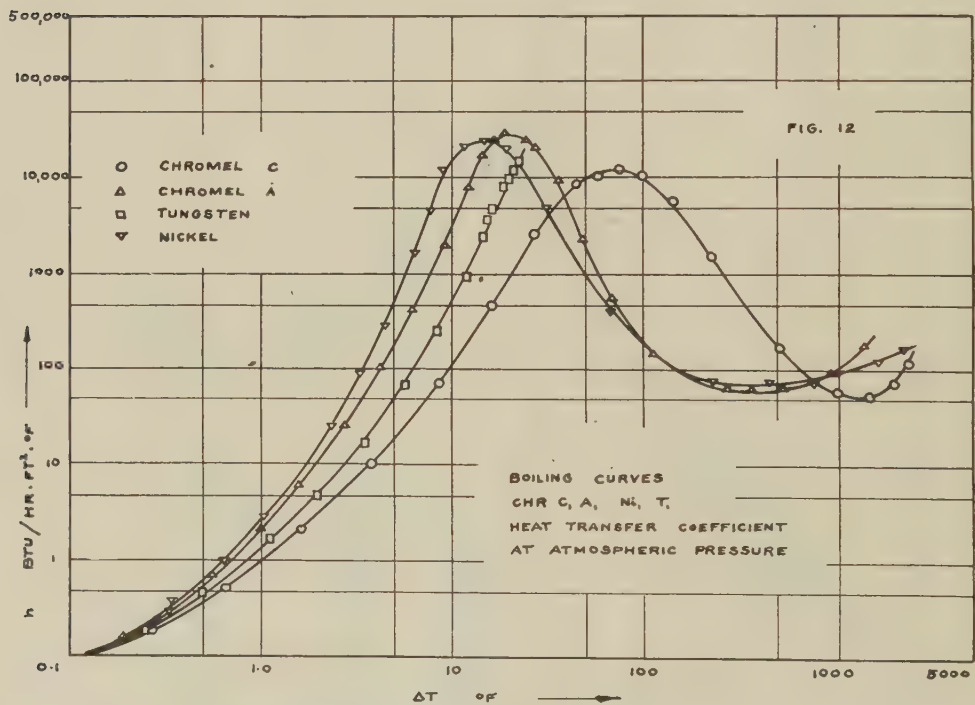


FIG. 12 COMPARISON OF BOILING CURVES OF CHROMEL C, CHROMEL A, NICKEL, AND TUNGSTEN TEST WIRES

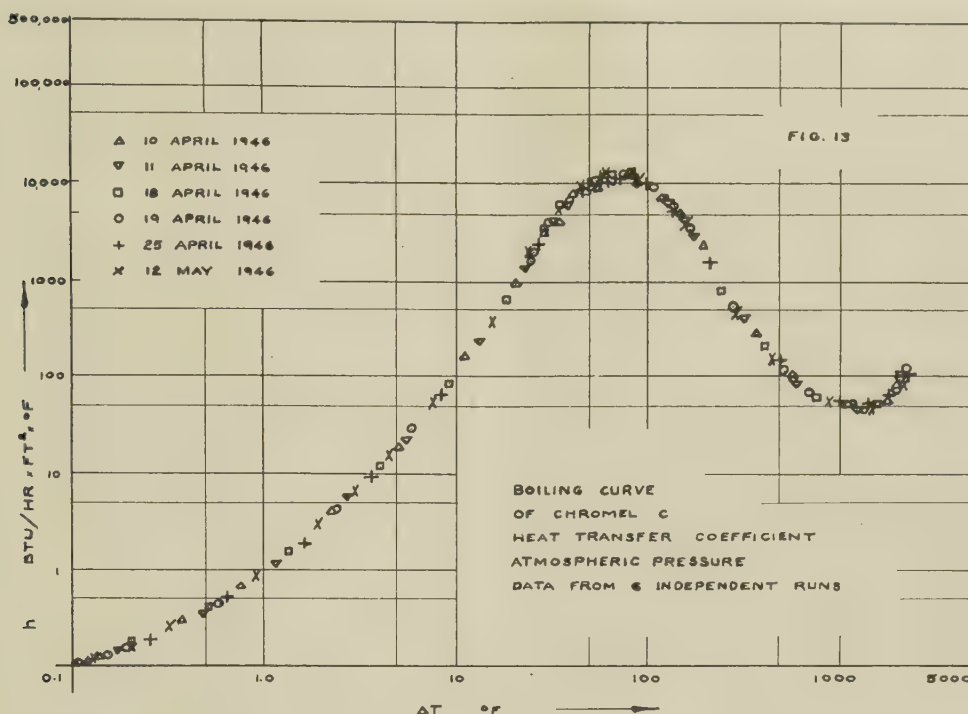


FIG. 13 COMPARISON OF SIX TEST RUNS WITH DIFFERENT WIRES ON DIFFERENT DAYS

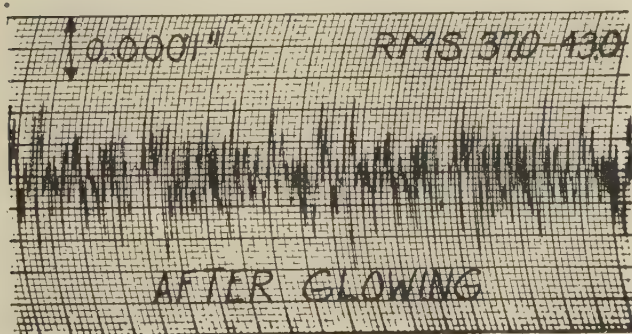
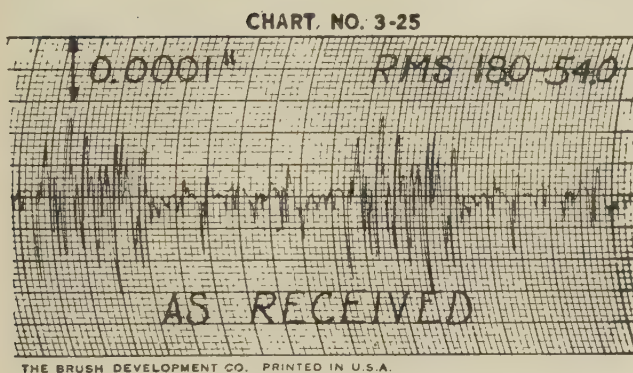


FIG. 14 SURFACE PROFILES MADE WITH SURFACE ANALYZER

the more fine-grained character of that roughness, as shown by the profile trace, was thought to be a contributing factor in explaining the reproducibility of data after the wire had been made to glow under water.

The available roughness measurements have suggested as a working hypothesis that the more uniform and fine-detailed roughness of a wire surface, produced by heating the wire to

incandescence under water, largely accounts for the reproducibility of the heat-transfer data from such wires.

CONCLUSIONS

A number of conclusions can be drawn from these preliminary experiments, as follows:

- 1 The boiling curve was found to be continuous over a wide range of metal surface temperatures, and the form of the curve remained the same. As the value of ΔT was increased from 0, the value of the heat-transfer coefficient increased to a maximum, then fell to a minimum, from which it increased steadily until the metal failed by melting.

- 2 At the same pressure, different metals gave different numerical values for the boiling curve.

At atmospheric pressure and a ΔT of 10, the heat-transfer coefficient was 100 for chromel C, 500 for tungsten, 3300 for chromel A, and 20,000 for nickel.

- 3 At different elevated pressures, the same metal gave different numerical values for the boiling curve.

For chromel C and a ΔT of 10, the heat-transfer coefficient was 100 for 0 psig, 1200 for 50 psig, 8000 for 75 psig, and 90,000 for 100 psig.

- 4 The mechanism of the boiling process is different for different parts of the boiling curve.

The boiling process at atmospheric and elevated pressures was observed, described, and illustrated in the paper.

- 5 The mechanism of the boiling process permits the same heat-transfer rate with three different values of ΔT .

For chromel C at atmospheric pressure, a heat-transfer rate of 100,000 Btu/(hr)(ft²) was allowed at ΔT values of 30, 430, and 1650 F, the corresponding values of the heat-transfer coefficient being 3330, 232, and 60.5. The presence of such boiling conditions may explain certain industrial operating phenomena.

- 6 The electrically heated, submerged-wire method of studying the boiling process is promising, and its further use and de-

velopment should be encouraged. The precision of the surface temperature measurements should be increased.

ACKNOWLEDGMENT

These experiments are part of a research project in the University of Missouri Engineering Experiment Station and were begun in 1937.

The authors are indebted to Mr. John Thurlo for assistance in the design of the elevated-pressure apparatus, and to Mr. Paul Landtiser for the water control. The authors are most grateful to Mr. Paul Ogden for his measurements of wire surface finish and his photomicrographs of wire sections.

BIBLIOGRAPHY

- 1 "Heat Transmission," by W. H. McAdams, McGraw-Hill Book Company, Inc., New York, N. Y., second edition, 1942, pp. 294-339.
- 2 "The Basic Laws and Data of Heat Transmission," by W. J. King, *Mechanical Engineering*, vol. 54, 1932, pp. 560-563.
- 3 "Heat Transfer in Evaporation and Condensation," by Max Jakob, *Mechanical Engineering*, vol. 58, 1936, pp. 643-660.
- 4 "Boiling," by T. B. Drew and A. C. Mueller, *Trans. AICHE*, vol. 33, 1937, pp. 449-473.
- 5 "Maximum and Minimum Values of Heat Transmission From Metal to Boiling Water Under Atmospheric Pressure," by S. Nukiyama, *Journal of the Society of Mechanical Engineers, Japan*, vol. 36, June, 1934, pp. 367-374. Abstract in English, *ibid.*, pp. S53-S54.

Discussion

R. C. COREY.⁴ The results obtained by the authors for the variation of the heat-transfer coefficient, h , as a function of the temperature gradient between the heater and the liquid, are extremely interesting and thought-provoking. That heat-transfer rates as high as 1,000,000 Btu per hr and sq ft could be attained with temperature differences of the order of 10 deg F, as noted for chromel C at a gage pressure of 100 psi, suggests lines for fundamental research on the heat-transfer characteristics of the steaming surfaces of boiler tubes at pressures up to 2000 psi. To date, estimates of the temperature gradient between the fluid and the tube metal in furnace wall tubes absorbing heat at rates of the order of 100,000 Btu per hr and sq ft are based upon assumed film coefficients between 5000 and 20,000, and therefore are subject to considerable controversy.

The authors mention that the wire temperatures were determined independently by means of thermocouples and an optical pyrometer, and that at 1500 F, the two methods agreed within 30 deg F. However, they concede inherent errors in the thermocouple method but neglect to state that measurements with the optical pyrometer also may be seriously in error, depending upon whether the radiator is surrounded by a hotter or a colder environment and upon the temperature and emissivity of the wire.⁵ Since the interpretation of the results depends to some extent upon accurate determinations of the temperature gradient, particularly in the regions where the minimum occurs in the h -values, it would seem desirable to investigate more thoroughly the errors in the temperature measurements.

It is stated in the paper that in zone II, "... many small spheroidal bubbles would leave the metal surface, some combined to form larger bubbles, all condensed in superheated liquid before reaching the liquid-vapor interface." It is not clear to the

writer why the authors assume the liquid to be superheated. Condensation of the bubbles would occur if the temperature of the liquid were not uniform.

Since the h -value for given temperature gradients was influenced markedly by the composition of the heater wire, and the maximum value was a function of pressure, some explanation for the abrupt cutoff for tungsten wire, which is shown in Fig. 12, would be desirable.

The marked difference found for the heat-transfer characteristics of chromel A, chromel C, and nickel wires is of considerable interest. Larson's work⁶ with metallic ebullators, in which he studied the phenomena of liquid superheat and of nucleate boiling in connection with heat transfer, suggests that the important criterion of good ebullators is nonwettability and that physical roughness or porosity of the surface did not appear to be a factor. Since he found marked variations in superheat for different metals and alloys, it seems likely that the nature of the surface as regards the valence type and the crystal structure of the phase comprising the surface may influence strongly the adhesion-free energy of the surface. In the authors' experiments, the marked differences found between chromel A and chromel C, which are essentially chromium-nickel alloys, leads one to suspect that pretreatment of the wires under water, to secure reproducible results, resulted in oxide phases of different compositions for the two alloys. It is well known that oxidizing treatment of single-phase chromium alloys leads to diffusion of the elements to the surface at different rates, with the result that the ratio of the metal ions in the surface oxide may be quite different from that of the parent metal.⁷

W. H. McADAMS.⁸ This paper presents data over a wide range of temperature differences for water boiling on four different metal surfaces. It represents a substantial contribution to the literature in the rather poorly understood field of heat transfer to boiling liquids, both at atmospheric pressure and elevated pressures.

In view of the interest in this field, it is hoped that the authors will publish the original data and not merely plots of coefficients versus temperature difference. It is difficult to read these plotted values with reasonable precision, in view of the number of logarithmic cycles required in the graphs. It is believed that other workers would appreciate complete tables of data, showing run number, wire material, wire diameter and length, location of voltage taps, power dissipated from wire, type of electricity (d c or a c in each case), location of thermocouples on each wire, pressure, temperature of the liquid, whether or not fouling or discoloration of the wires occurred, and readings by optical pyrometer (when used).

The significance of readings of thermocouples attached directly to a cylinder carrying either a-c or d-c electricity, has long been a matter of conjecture, since the drop in applied voltage across the width of the thermocouple might not be negligible to the voltage set up by the thermocouple.

Surely some heat is conducted from the power wire along the leads of the thermocouples and dissipated to the liquid, causing the heat-flux density and coefficient, based only upon the area of the power wire, to be too high. When the electrically heated wire was quite hot, was any boiling noted from the leads of the thermocouple? Even though temperatures measured by a thermocouple agreed within 30 deg F with those obtained by

⁴ Supervising Engineer, Combustion Research Section, Central Experiment Station, Bureau of Mines, Pittsburgh, Pa. Published by permission of Director, Bureau of Mines, U. S. Department of the Interior.

⁵ "Errors in Temperature Measurement by Radiometric Measurements," by W. T. Reid and R. C. Corey, *Combustion*, vol. 15, February, 1944, pp. 30-34.

⁶ "Factors Affecting Boiling in a Liquid," by R. F. Larson, *Industrial and Engineering Chemistry*, vol. 37, 1945, pp. 1004-1009.

⁷ "Corrosion by Hot Gases," by R. C. Corey, *Combustion*, vol. 15, November, 1943, pp. 34-39.

⁸ Professor, Department of Chemical Engineering, The Massachusetts Institute of Technology, Cambridge, Mass.

the optical pyrometer, if the latter were sighted on the thermocouple, errors due to conduction in the leads might still be substantial, since both measured temperatures might be significantly lower than that of the wire with no thermocouple attached. Would the authors care to give a quantitative estimate of the errors caused by conduction in the leads, and to report any measured gradients in temperature along the test section of the power wire?

Because of the possibility of error in temperature measurement, it would be safer to plot heat-flux density versus temperature, rather than coefficient versus temperature in order to isolate any possible error in the abscissa.

E. P. PARTRIDGE.⁹ Any attempt to correlate what happens at the surface of a horizontal electrically heated wire 6 in. long and 0.04 in. diam with what happens on the internal surface of a vertical boiler tube 50 ft long with an internal diameter of 0.92 in. must seem a mere exercise of the imagination. Nevertheless, the film boiling produced in the laboratory by the authors of this paper differs only in magnitude and in detail from the film boiling observed by Davidson, Hardie, Humphreys, Markson, Mumford, and Ravese, published in their important paper five years ago.¹⁰

Fig. 30 of that paper indicates the existence of a "hot spot" part way up the vertical 50-ft wall tube, with normal temperatures above and below this limited region of overheating. In their discussion of this figure, Davidson, et al, say, "The value of Δt rose above the limits of the chart shortly after point 10 and returned to normal values at point 16. It is in this length of the tube that steam-blanketing or the co-existence of continuous phases of vapor and liquid is assumed."

The present authors appear to have duplicated, under laboratory conditions, the creation of a localized region of film boiling. Even though they heated a portion of their experimental wire to incandescence to produce this condition, the temperature of the metal surface may not have been much higher than that indicated by the microstructure of some boiler tubes which have failed in service at localized hot spots.

F. B. SCHNEIDER.¹¹ In so far as the test arrangements are concerned, one of the most important measurements is the determination of the temperature of the wire surface and of the mean temperature of the water. It is believed that if a typical test reading were published it would help to answer some of the questions in the mind of the reader. In the absence of this information, the writer would be interested to know the answers to several questions.

Every precaution has been taken to assemble the thermocouples correctly to the wire, according to the best-known practice. They were attached at right angles to the wire and at various positions around the perimeter. It was found that such a location does not cause a variation of the temperature. This agrees with previous experiments in so far as higher temperature differences are concerned. However, at lower temperature differences between metal and water, substantial variations of the wire temperature at various positions around the perimeter have been observed in previous tests with tubes of $1/2$ in. diam and larger. Since the wires of this investigation had a diameter of 0.040 in., it would be valuable to know for what diameters the influence of the location of thermocouples can be noticed.

⁹ Director of Research, Hall Laboratories, Inc., Pittsburgh, Pa.

¹⁰ "Studies in Heat Transmission Through Boiler Tubing at Pressures From 500 to 3300 Pounds," by W. F. Davidson, P. H. Hardie, C. G. R. Humphreys, A. A. Markson, A. R. Mumford, and T. Ravese, *Trans. ASME*, vol. 65, 1943, pp. 553-591.

¹¹ Locomotive Engineering Division, General Electric Company, Erie, Pa. *Mem. ASME*.

The presence of the thermocouple wires causes deflection of the circulation currents around the wire which has led to misleading results in previous experiments. From the illustrations and from the description it is not clear where the thermocouples were fastened longitudinally. The writer believes that the correct location would have been at the center of the wire, in order to avoid the cooling effect of the bus bars and of the walls. However, special tests would have to be carried out to determine the effect of circulatory disturbances. In this investigation the readings of the thermocouples were checked with thermometers only at noncirculating flow.

The mean water temperature was measured by a thermometer suspended above the center of the horizontal wire, as shown in Fig. 1 of the paper. Since this will measure the temperature of the warm current of water and of bubbles rising from the wire before the current is dispersed by the circulatory motion, the recorded temperature difference between wire and water will be a minimum. If the thermometer was located above the center of the wire, but laterally displaced toward the wall, probably higher temperature differences would be obtained. The difference in readings may be important at low values of ΔT . The distance between the thermometer and the wire is also important since the temperature of the rising current varies with height. Therefore the writer would appreciate any information about temperature traverses in the vessel during the tests.

The influence of the type of water used in the test is considerable. With the use of distilled water, the heat-transfer coefficient may be 25 per cent lower than with contaminated water. Also, a film of oil on the wire may reduce the heat-transfer values considerably within the temperature range of tests previously taken. In this investigation the wires were subjected to glow for 15 min before the tests were started. This may have caused a thermal decomposition of the oil with an accompanying increase of heat transfer due to scaling. If tests have been undertaken to determine the influence of distilled water, water with special agents, wires free of oil before glowing, and wires with an oil film before glowing, the writer would be indebted to the authors for publishing them.

From the "Visual Observations" as described in item II and shown in sketch II of Fig. 7 of the paper, we learn that small spheroidal bubbles combine to form larger bubbles. Previous publications of similar observations maintain that the bubbles grow rapidly from small into larger ones. According to McAdams, the superheat is consumed in vaporizing additional liquid, thus increasing the volume of the bubble. He substantiates his statement by referring to the stroboscopic photographs of Jacob and Fritz who measured the increase in volume of typical bubbles. They obtained an increase of up to 4500 per cent of the original volume.

In addition, the writer cannot agree with the authors that the larger bubbles condense in the superheated liquid before reaching the liquid-vapor interphase. Since the heat transfer into the liquid is considerably higher than into vapor, the temperature of the vapor in the bubble, even if superheated, must be lower than the temperature of the liquid. This temperature gradient permits the evaporation of additional liquid whereby the remaining part could be cooled to the temperature of the vapor in the bubbles. Then this process is terminated. The bubbles attained maximum size. In order to condense the vapor in the bubbles completely, the temperature of the surrounding liquid must be below the saturation temperature of the vapor. Therefore in the region where the bubbles disappear the liquid cannot be superheated. On the contrary, the temperature of the liquid must be subcooled. Since no other heat loss can occur, the only explanation is the mixing of the previously superheated liquid with cooler liquid in the apparatus which did not pass the hot wire. At this stage of the boiling process, the location of

the thermometer which determines the mean temperature of the water is important. At higher values of heat dissipation, the total amount of water in the vessel becomes superheated with the result that condensation of vapor is impossible. The bubbles reach the liquid-vapor interface of the liquid as described in item III.

In Fig. 8 of the paper the authors describe the very interesting formation of a funnel-shaped film and observe that no bubbles appear close to the glowing wire. In one section the authors remark that the wire was heated to incandescence. If this means the radiation of white light, the temperature of the wire must have been above 1500 F, which agrees with the authors' statement that ΔT was above 1000 F in sketch VI, Fig. 7. If these temperatures were above 2500 F, the dissociation temperature between hydrogen and oxygen, it is possible that the film around the wire, as shown in Fig. 8, did not consist entirely of steam, but also of hydrogen and oxygen. The high velocity of the bubbles and the noise may indicate explosive combustion of the hydrogen-oxygen-gas mixture.

The data derived by the authors and illustrated in Figs. 9 to 12 of the paper can be compared with similar information in the region of atmospheric pressure only. The test results of this investigation furnish film coefficients of heat transfer of 11,000 to 30,000 Btu per hr sq ft deg F for chromel C and A at critical temperature differences of from 20 deg F to 70 deg F. The authors found a maximum film coefficient of 25,000 Btu per hr sq ft deg F at a critical temperature of 30 deg F for nickel. McAdams published values for chrome-plated copper of from 3300 to 8600 Btu per hr sq ft deg F at 45 deg F, and for nickel-plated copper of from 3800 to 8200 Btu per hr sq ft deg F at 50 F to 42 F at atmospheric pressure. Chromel C attained approximately the same film coefficients as the maximum values reached for chrome-plated copper. However, chromel A and nickel show values which are far above similar results. On the other hand, the critical temperature difference for chromel C is 70 deg F, while the highest critical temperature difference previously known for metal in water is about 55 deg F.

Nickel and chromel A show a critical temperature difference of approximately 20 deg F, which is one half of the hitherto known value. Since the maximum values of the film coefficients found by the authors for chromel A and nickel are very high, multiples of those for chromel C, copper and steel, the writer would appreciate knowing upon what characteristics of the materials or the liquid the authors base their explanation for this outstanding result. A duplication of their findings would be extremely valuable in commercial evaporative processes.

On the other hand, at a temperature difference of 5 deg F, the values of the film coefficients of heat transfer at atmospheric pressure are from 20 to 500 Btu per hr sq ft deg F while previous tests furnish data of 220 to 700 Btu per hr sq ft deg F. About 220 seemed to be normal. Therefore it is remarkable that chromel C and tungsten have such low coefficients as 20 and 50 Btu per hr sq ft deg F.

In conclusions 1 and 2 of the paper, the authors describe the standard boiling curve. It would have been valuable to substantiate by test that this curve can be straightened into a flat line from zero ΔT to critical ΔT at the maximum heat-transfer coefficient with sufficient agitation at both atmospheric and elevated pressures. Furthermore, it is not known if this line extends over the critical temperature and continues until it meets the rising branch of the boiling curve (see VI in Fig. 6). In conclusions 3, the authors furnish data of the remarkable increase of heat transfer by increasing the pressure to 50 or 100 psi.

As a detail matter, the writer would like to mention that at $\Delta T = 10$, $h = 500$ for tungsten as per Fig. 12, and not 155 as described.

In concluding the discussion, the writer believes that the industrial world connected with evaporative processes is indebted to the authors for their important findings which may well stimulate further research in this particular field. Although there are many questions which remain unsolved, the results shown in this paper can be used immediately and to great advantage in commercial applications.

B. J. CROSS¹² AND W. S. PATTERSON.¹³ Research work such as that reported by the authors is a valuable contribution to the technical literature when it confirms the work of earlier investigators or extends the investigation into unexplored regions as the authors have done by working up to 100 psi with extremely high rates of heat flow. Such work is particularly valuable if the results lead to conclusions and recommendations for application to industrial-equipment design problems. However, the writers would like to have the authors explain the following:

1 The authors present data which show that different metals give different boiling curves and different film coefficients. Is this not more a function of surface condition than a function of the composition of the wire? It is stated that for any wire the results could not be duplicated until the wire was first conditioned by heating to redness when immersed. Was the surface finish of all test wires identical before and after use for the boiling tests? What were the "visual and thermal inspections" which led to the rejection of 9 out of 10 test-wire assemblies?

It is known that practically all metals are chemically attacked by water. Therefore it is reasonable to expect that the four metals used in the heating elements would each be differently attacked and each would have a different surface condition.

It would also be expected that mild steel, the material most commonly used in commercial heat-transfer apparatus, would give other values for film coefficient than those reported.

If this suggested explanation of the difference in the results with the four wires used is true, then the composition of the water would be a factor. Has this been given consideration?

2 Could the condensation of steam bubbles as described in item II of "Visual Observations" be due to heat losses from the pressure vessel, which would result in nonuniform water temperature throughout the vessel, particularly at low rates of boiling? It seems difficult to account for bubble condensation by the method suggested by the authors.

3 The authors state that circulation was less at elevated pressure compared to atmospheric pressure. Does this refer to observed velocity of the bubbles flowing upward from the wires or water velocity approaching or passing the test wire? That less circulation produced thinner films is contrary to what we would expect.

We agree with the authors that their method of studying the boiling process is promising, and its further use and development should be encouraged. The use of a single horizontal wire may, however, give results which are different from what would be obtained with a bank of horizontal wires or tubes (to simulate a commercial evaporator). In the latter case, the circulation conditions would be different in many respects depending upon tube arrangement and width of lanes between tubes. Evaporation from the inside or outside of vertical tubes could be simulated by mounting the test wire vertically and enclosing it in a tube to force the steam generated to increase velocity as it approached the top of the test wire, thereby duplicating to a certain extent conditions which exist in vertical boiler tubes.

Some relationships pertaining to maximum values of Δt , h , and

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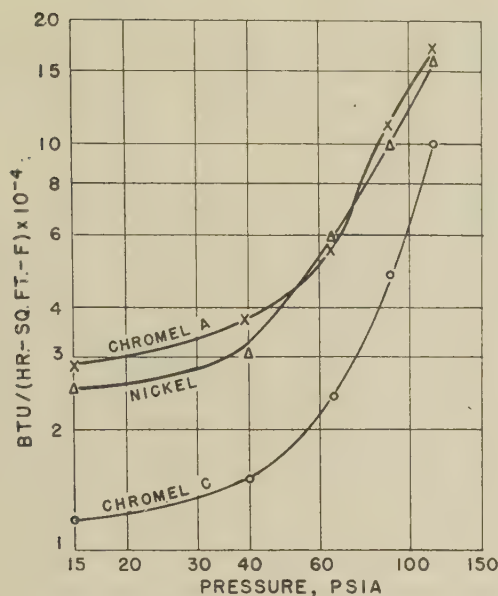


FIG. 15

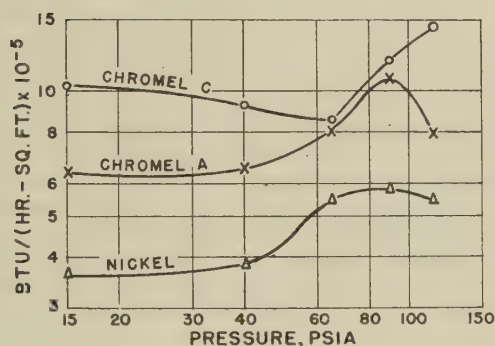


FIG. 17

q/A are plotted in Figs. 15 to 18, inclusive, of this discussion, using data from the authors' curves. Although the shape of the curves for h maximum, Fig. 15, and Δt at maximum h , Fig. 16, are quite similar for the three metals when plotted against pressure, the curves for q/A at maximum h (Fig. 17), are not similar. For two of the metals, a maximum point appears at 90 psi, above which the total heat flux drops off. The same phenomenon is illustrated when q/A maximum is plotted against pressure, Fig. 18. Do the authors have any explanation for these phenomena or for the fact that their data show more than twice the maximum heat flux for chromel C than for nickel heating surface?

It is interesting to note that heat-transfer rates in excess of 1,000,000 Btu per hr per sq ft were obtained with nucleate boiling and with temperature differences of the order of 10 to 20 deg F between boiling surface and fluid.

AUTHORS' CLOSURE

This paper was accorded encouraging and thoughtful discussion at Chicago, and the foregoing discussers have submitted their remarks for publication. The authors thank all discussers for their interest in the subject and their helpful comments.

The manuscript has been extended for publication to include the original data in accordance with Professor McAdams' suggestion, and other items of general interest have been added. Since the questions raised by the discussers are quite similar, our reply has been directed to those topics.

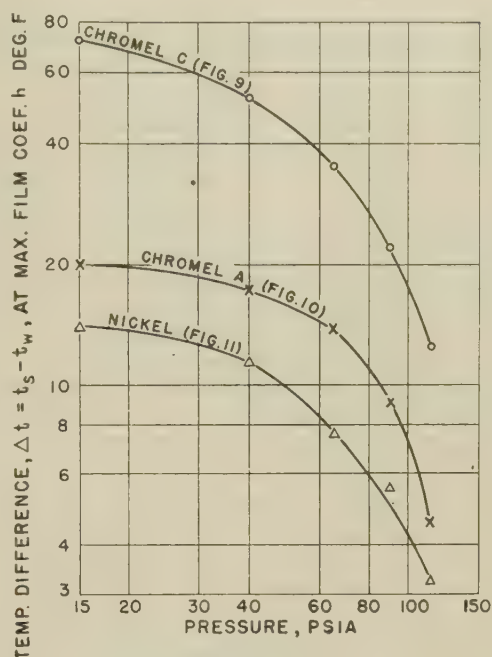


FIG. 16

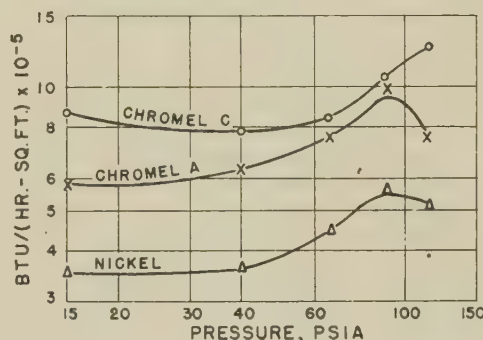


FIG. 18

Condensation of Zone II Bubbles

The type II boiling mechanism was observed in the open jar shown in Fig. 1 and in the test apparatus shown in Fig. 2. The test apparatus was very heavily insulated, and by means of the electric immersion heater in the vertical header, the whole water content of the apparatus was brought slowly to a streaming condition at the desired test pressure. The heat loss was so small that tests could be made without operating the immersion heater. The thermometer well used in the measurement of the water temperature was located about midway between the experimental wire in the pipe-cross and the vertical header containing the steam space. Since the water temperature checked the saturation temperature corresponding to the pressure, it was concluded that the small currents of superheated liquid steaming upward from the test wire to the liquid-vapor interface in the header had been well mixed before passing the thermometer well.

When operating at elevated pressures, a novel check on the absence of subcooled liquid in the test section of the pipe-cross was available. Strong illumination was arranged at one sight glass so as to floodlight the whole test section as observed from the opposite sight glass. With the test wire not operating, the saturated water was clear and free of steam bubbles and striations. Then by valving off steam, the pressure was reduced until the first vapor bubbles appeared. These bubbles were of the nucleate

type formed at favored spots. The observed even character of the distribution of these bubbles throughout the interior surfaces of the test section was considered strong evidence against the presence of undercooled surfaces and pockets of subcooled water.

Much closer inspection of the boiling process was possible when using the open jar of Fig. 1, and many hours have been spent in such observations. To insure against the presence of subcooled liquid, eight or more thermometers were placed throughout the water, and the jar was covered with insulation in which inspection doors were provided. Under these conditions, the authors and their associates have often observed the type II boiling mechanism with the same results. The bubbles were originated at favored spots and with few exceptions were very small, suggesting a mist. The tendency for these bubbles to combine before condensing was later observed to be less frequent than at first believed. Condensation of the zone II bubbles was always completed within a few inches of the test wire.

Temperature Measurements

The water temperature was influenced by the location of the thermometer. Referring to Fig. 1, for locations directly above the test wire, the temperature was somewhat higher due to the presence of superheated liquid in this region. Quite uniform water temperatures were observed at locations to the side and below the test wire. In the test apparatus shown in Fig. 2, the thermometer well was located at a point 16 in. distant horizontally from the center of the test wire in order to minimize the influence of superheat on the observed water temperature.

Regarding the test wire thermocouples, no evidence of boiling was observed along the thermocouple wires. With the limited facilities of our laboratory, the construction of a test wire and thermocouple assembly was so difficult for our students that no attempt was made to further complicate the construction by fastening secondary thermocouples to the wires of the primary thermocouples in order to observe the operating gradient.

As to the effect of test-wire diameters above 0.040 in. on the uniformity of the surface temperature around the wire, some experiments have been made with 0.102-in.-diam wires and very nearly uniform temperatures were observed around the wire.

Some error must have been present in the optical-pyrometer measurements of the surface temperature. The wire was assumed to be a hot body surrounded by a colder body, and the emissivity of the wire surface was taken as equivalent to that of solid iron oxide. These conditions are covered by Table 1 of an excellent paper⁵ by Reid and Corey. From this table, an optical-pyrometer observation of 1600 F corresponds to a true surface temperature of 1603 F, a small error. However, our assumptions have not been proved, and one of our graduate students, D. E. Hall, has undertaken an experimental study of the problem.

Test Wires and Test Water

The Farber method¹⁴ of standardizing a metal surface for boiling consists of heating the metal component to incandescence for a short period of time while submerged in the boiling liquid. After applying the treatment to a test wire, the surface was covered with a dark-gray coating or scale, very thin and tough. Once formed, this new surface appeared to be quite stable or at least subject only to a very slow rate of growth. When examined at various magnifications, the surface was observed to have a uniform, lacelike, or spongy structure. The gas film which covered the surface during the process was composed of steam and its dissociation products. Considering the materials present, the lacelike fine structure of the surface was very probably covered by a thin film of oxides.

¹⁴ "Structural Changes in a Chrome-Nickel-Iron Alloy Due to High Temperature Heat Flux," by Paul Ogden, J. T. Kimbrell, and R. L. Scoria, NEPA Symposium on Heat Transfer, Oak Ridge, Tenn., December 8-12, 1947.

The Farber treatment was applied to wires in the "as received" condition, free from ordinary dirt. The effect of preliminary coatings of oil and other agents was not investigated. In the case of tungsten, it was not possible to secure incandescence as the wire failed by chemical action when ΔT exceeded 21 F at atmospheric pressure.

The over-all result of the Farber treatment was considered to be the establishment of more durable and more uniform physical and chemical properties at the solid-fluid interface, the principal properties being the surface tension, emissivity, and chemical stability of the solid surface. The Farber treatment was able to make the surface properties more uniform for each kind of wire but was not able to make the surface properties more alike for various kinds of wire. The characteristic differences in the interface properties of different metal-water combinations persist after the Farber treatment and account for the difference in the boiling performance of different kinds of wires, as shown in Fig. 12 and Figs. 15 through 18. The fundamental phenomena involved here were the subject of an important paper published recently by Professor Larson.⁶

The test water was required to be clear for good visual observation of the test wire and to operate without fouling the wire surface. In the open jar of Fig. 1, where solid-nickel bus bars were used, distilled water was employed. In the test apparatus of Fig. 2 distilled water discolored rapidly. Various agents such as trisodium phosphate and sodium hydroxide were added to adjust the pH value of the solution to the point of minimum solubility for iron. The results were not altogether satisfactory. The procedure finally adopted was to wash and boil the apparatus with the immersion heater using a pH = 10 solution of distilled water and sodium hydroxide. The solution was then drained, the sight glasses cleaned, the test wire assembly installed, the apparatus filled with distilled water and brought to a steaming condition at the test pressure. For a considerable period of time the test water remained clear and satisfactory for tests and visual observations.

Applications

Imagination like the spirit is unquenchable, and there must be some correlation between a laboratory test wire and a 50-ft boiler wall tube. Aside from geometrical and material differences, the heatflux and fluid dynamic effects were uniform along the test wire and variable along the tube.

The open jar of Fig. 1 has been the scene of many interesting experiments intended to simulate some feature of commercial boiling equipment. Our students have made resistance elements of many different materials and shapes and have observed the resulting effects. For example, when the natural circulation currents were confined to flow through a closed passage made of glass walls, the circulation velocity increased, throwing scalding water out of the jar, and the passage above the heated wire eventually became steam-bound. This simple step in changing the original laboratory conditions toward those simulating the conditions of commercial apparatus seriously complicated the whole boiling process. For these reasons the present tests were directed toward a study of boiling under fundamentally simple conditions.

ACKNOWLEDGMENT

The authors thank Mr. E. P. Partridge for calling attention to the subject of hot spots in wall tubes, and thank Mr. F. B. Schneider for correcting the statement in conclusion 2 regarding the coefficient for tungsten at $\Delta T = 10$ where $h = 500$ and not 155.

The authors are deeply indebted to Mr. J. T. Kimbrell, who succeeds Mr. Farber in this laboratory, for his skillful production of the color film used in the presentation of this paper.

Radio-Frequency Heating in the Woodworking Industry

By F. L. MACALUSO,¹ BALTIMORE, MD.

Radio-frequency heating is finding wide application in the woodworking industry for such processes as edge-gluing, plywood manufacture, drying, assembly-gluing. The theory of radio-frequency heating is reviewed briefly, and how this type of heat is applied in various applications is described. Several case histories of actual manufacturing problems are taken up. The recent Federal Communications Commission Regulations on the use of radio-frequency power for industrial heating are treated comprehensively.

RADIO-frequency power is being used in the woodworking industry to produce better products cheaper and faster.

While the advent of synthetic-resin glues made possible the manufacture of stronger, more durable products, the cheaper and more rapid results obtained from using these glues were not fully realized until a tool was developed for setting these resins quickly and efficiently. This tool is the industrial radio-frequency generator which is being used economically in industry for a variety of purposes including edge-gluing, plywood manufacture, assembly-gluing of cabinets, manufacture of timbers and spars, lamination of tennis-racket frames, and drying.

THEORY OF RADIO-FREQUENCY HEATING

Heating of wood or other "nonconducting" material by radio-frequency power is not mysterious and is essentially the same as electrical heating of any other material. The basic heat equation is

Btu = pounds heated \times specific heat \times temperature change, deg F. [1]

Converting this to electrical power units

$$Kw = 1.76 \times 10^{-2} Mc(T_2 - T_1) \dots \dots \dots [2]$$

where

M = lb per min heated

c = specific heat

$T_2 - T_1$ = temperature change, deg F

Using Equation [2], the powers required to heat 1 lb of copper, steel, and wood through 100 F in 1 min are 162 w, 208 w, and 616 w, respectively, the difference in power requirements being due entirely to the difference in specific heats of the three materials.

Electrical heating is accomplished by passing a current through the piece to be heated, and the value of this current squared times the resistance of the piece is the power. Considering a pound of copper, a cube about 1.46 in. on the side, the voltage and current required are approximately 0.0087 volt and 18,650 amp. This illustrates why copper is used as an electrical conductor

and other means are used for heating it. To heat 1 lb of nichrome steel (high electrical resistance), a cube 1.5 in. on the side, the voltage and current required are approximately 0.074 volt and 2820 amp. While the current is still high, it is considerably less than for the copper cube and if drawn out into not too fine a wire would heat readily with not too great a current.

To heat 1 lb of wood, a 1 \times 6 board 8 in. long, with 60-cycle current and with electrodes on the 6 \times 8 faces requires approximately 2,250,000 volts, and 0.027 amp. This high voltage is extremely impractical. It is calculated using the fundamental dielectric heating equation

$$Pv = 1.4 E_1^2 f e'' \dots \dots \dots [3]$$

where

Pv = power density, watts per cu in.

E_1 = voltage gradient, kv per in.

f = frequency in megacycles

e'' = loss factor, a property of the material which varies somewhat with frequency

Using a radio frequency of 13.6 megacycles this heating can be accomplished with 2100 volts and 5.8 amp, which are entirely reasonable figures for RF heating. The only justification for radio frequency in the dielectric heating of nonconducting materials is to permit heating with reasonable voltages.

PLYWOOD MANUFACTURE

One of the first, commercially successful applications of RF heating in the woodworking industry was in the manufacture of plywood. In this process, the sheets of veneer spread with glue are placed between large flat electrodes in a press. The electric field between electrodes is perpendicular to the glue lines and, since the electrical properties of the glue are such that it heats more slowly than the wood, the glue is heated mostly by conduction from the greater mass of adjacent wood. The press platens may be cold, or they may be heated to prevent conduction losses from the wood into the press.

The problem easily resolves itself into heating the entire mass of wood to the setting temperature of the glue in a sufficiently long time to permit the glue to set and to permit the escape of steam without damage to the wood when heating must be carried over the boiling point. Ten sheets of 4-ft \times 8-ft \times 1/2-in. plywood can be heated from 70 to 140 F in approximately 10 min with a 20-kw RF generator. In RF heating, the heat is generated uniformly within the material and the whole board comes up to temperature practically simultaneously. When using a hot-platen press for thin sections, the time for the heat to penetrate to the remotest glue line is relatively short, while for thicker sections it is considerably longer and damage to the outer veneers may result. Hence there is a dividing line between favorable hot-platen-press applications and RF heating applications. Economic studies have shown that for plywood of 1/2 to 3/4 in. thickness or less, a multiple-opening hot-platen press is more economical than an RF generator. This dividing line varies considerably from plant to plant, depending upon the quantity of waste wood available for steam generation and upon local power costs. The dividing line will undoubtedly be lowered

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 47-A-115.

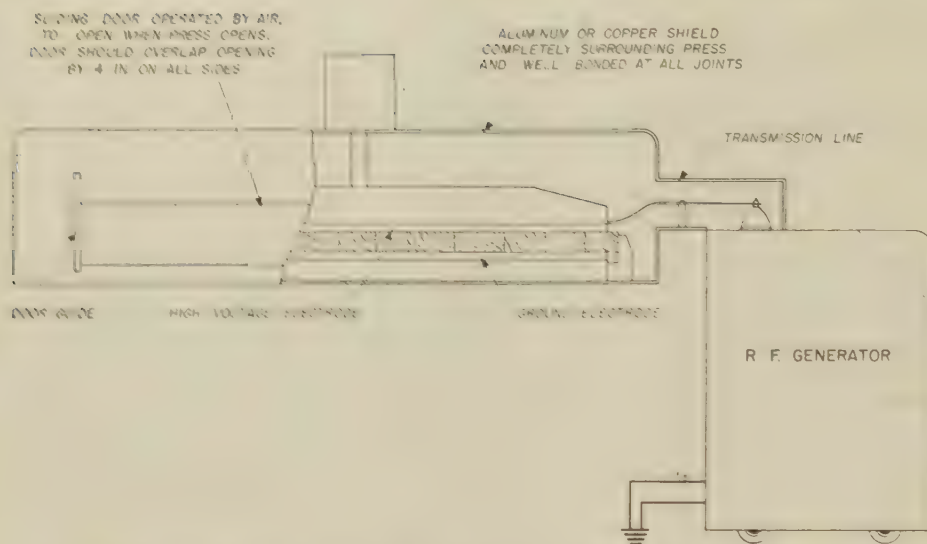


FIG. 1 DIAGRAMMATIC DRAWING OF EDGE-GLUING PRESS AND RADIO-FREQUENCY GENERATOR ASSEMBLY



FIG. 2 EARLE HART EDGE-GLUING PRESS WITH 10-KW RF GENERATOR

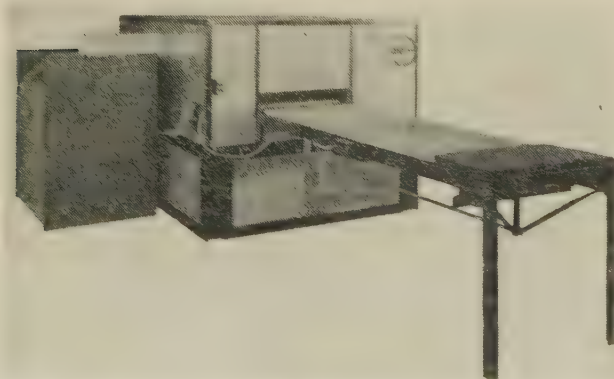


FIG. 3 WALDRON-FLEISCHBACH EDGE-GLUING PRESS WITH 10-KW RF GENERATOR

as RF generator and power costs are reduced and as the woodworking industry finds better uses for what is now waste wood.

The manufacture of curved plywood is considerably more favorable for RF heating than the manufacture of large flat panels, particularly on short runs, since curved wooden forms covered with thin metal electrodes are much more economical than curved metal forms with heating elements. A variety of curved plywood articles such as dresser-drawer fronts, laminated golf-club heads, radio-cabinet sections, and specialty furniture items are being produced economically with RF heating.

EDGE GLUING

The most important commercial application of RF heating in the woodworking industry today appears to be that of edge-gluing narrow boards to fabricate core stock or large open-face panels. The general practice is to glue several pieces from 2 to 4 in. wide and 1 in. thick into panels as large as 48 × 96. In the laminating process for making plywood, the electrodes are on the faces of the board, the electric field is perpendicular to the glue lines, and all the wood must be heated in order to heat and set the glue. In edge gluing, the electrodes are also on the faces of the board, but the glue lines are parallel to the electric field. Fortunately, a wet glue line has a much higher electrical conductivity and loss factor than the adjacent wood, resulting

in selective heating of the glue line with very little power going into the wood. An edge-gluing press-generator assembly is shown diagrammatically in Fig. 1, and Figs. 2 and 3 show examples of such installations. Here the individual boards are held flat between electrodes by a few pounds per square inch top pressure, and a 100 to 200-psi glue-line pressure is exerted from the sides.

For small panels on the order of 15 in. × 30 in., it has been common practice to use solid sheets of stainless steel for the electrodes, while for the larger sizes up to 48 in. × 96 in. either solid sheets or strips are used as electrodes. The construction and principle of strip electrodes is shown in Fig. 4. If the proper power and heating times are used, a glue line heated by strip electrodes will be set only along the zigzag pattern connecting the electrode strips. This highly selective heating makes for higher production and greater economy of RF equipment. When room-temperature-setting resin glues are used, the portion of the

glue not set by RF heat soon sets up, and 100 per cent glue-line strength is quickly developed. Using this process, 36-in. \times 96-in. boards made of 1-in. \times 3-in. pieces have been satisfactorily glued in 20 sec and less using a 10-kw RF generator.

In most plant operations, it is only necessary to develop sufficient glue-line strength to permit handling while full strength is developed during normal delays before other operations. Should greater initial strength be desired, it is only necessary to extend the heating time and the complete glue line will be cured. A further advantage of strip electrodes is that it simplifies generator load matching with the result that generator adjustments are never necessary except when changing sizes of assembled boards.

Although practically all RF heating for edge-gluing wood is done today with batch-type presses, a continuous RF edge-gluing press has recently been developed which tests indicated will produce at greater rates than the batch press. A schematic arrangement of the press is shown in Fig. 5, and Fig. 6 is such a press with its RF generator. The design of this press is mostly a mechanical problem of carrying the glued boards between electrodes and of holding the pieces firmly and free from vibration while the glue is being set. It can handle various widths of assembled boards, from 18 to 36 in., and of any desired length. The application of RF power on this press is a fairly simple problem, and in co-operation with RF generator manufacturers, the press components were designed and laid out to permit the most efficient utilization of RF power.

The use of RF power for edge-gluing affords particularly noteworthy savings, since it eliminates the cumbersome carrier clamps requiring many operators, saves considerable manufacturing and storage space, permits the use of high-temperature setting and highly water-resistant glues, permits a glue saving of as much as 50 per cent in some operations, and makes for greater uniformity with less rejects. A well-designed press generator installation is simple, small, clean, and easy to operate. Initial setup can be accomplished by a qualified technician in a few hours. Most installations have press and generator controls interlocked so that a single start button closes the press and applies RF power until the generator timer shuts off the power and opens the press.

FUNCTIONS AND PROPERTIES OF GLUE IN RF HEATING

The primary function of the glue is naturally to bond pieces of wood together. In applications such as through heating for plywood manufacture where the glue is heated by conduction from the wood, RF heating imposes no special requirements on the glue, and glue set by RF heat in this manner has the same physical properties, time-temperature setting curves, strength,

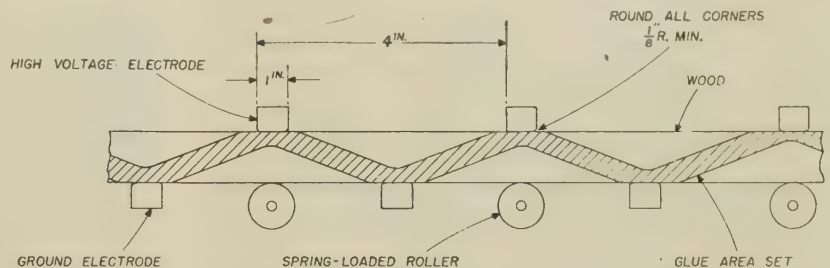


FIG. 4 STRIP ELECTRODE DESIGN FOR EDGE-GLUING PRESS

moisture resistance, etc., as the same glue set by other heating methods.

In edge gluing and in other processes, where the glue line is

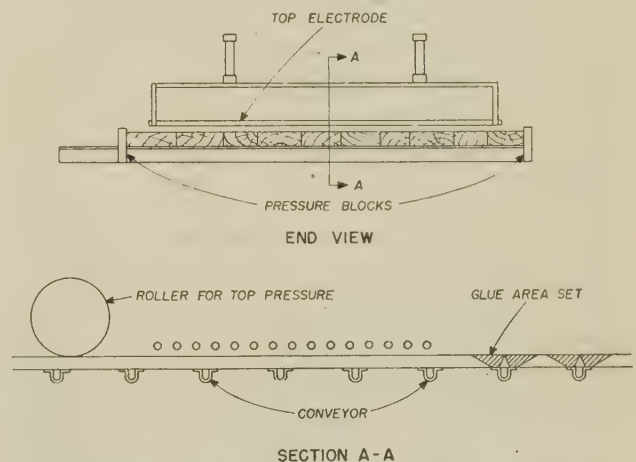


FIG. 5 DIAGRAM OF CONTINUOUS EDGE-GLUING PRESS

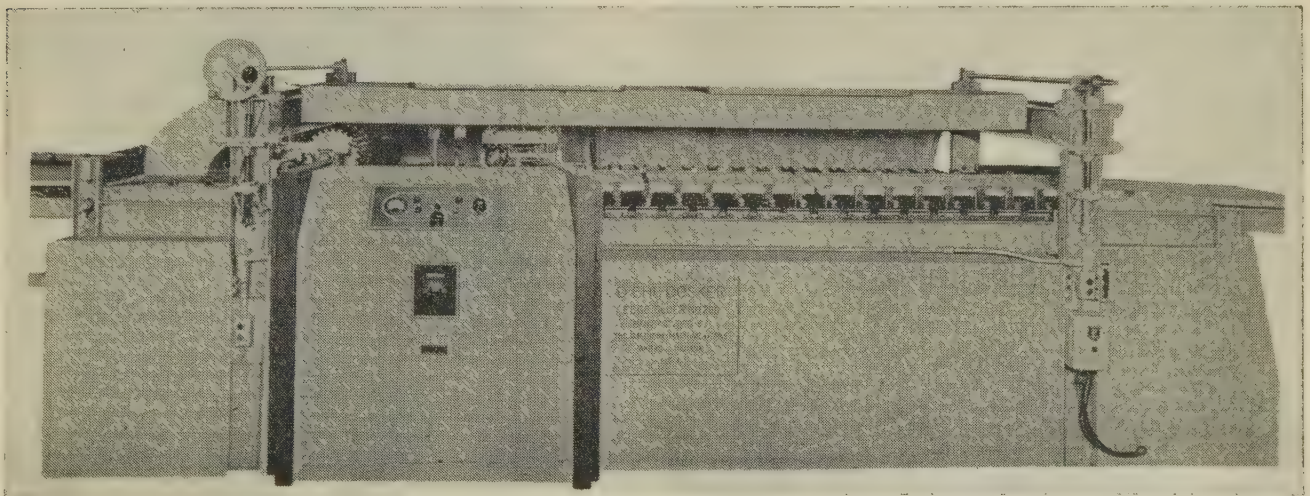


FIG. 6 DIEHL-DOSKER CONTINUOUS EDGE-GLUING PRESS WITH 10-KW RF GENERATOR

heated selectively by RF heat, the glue line is made to carry a considerable portion of the RF current and hence special properties are demanded. These properties are high conductivity, high loss factor, and high arc resistance. A number of resin glues are now on the market which are very satisfactory for this type of heating. In general, the urea, resorcinol, and melamine glues are satisfactory, while the alkaline phenolic glues are not. Some of these urea, resorcinol, and melamine glues were originally compounded for curing cycles of many minutes to many hours and, fortunately, have reacted satisfactorily with heating cycles of a few seconds. Considerable research is being done by glue manufacturers to adjust their resins to meet these new conditions, and greatly improved glues are expected.

SPECIAL GLUE-LINE-HEATING APPLICATIONS

Two specialized applications of glue-line heating or edge gluing are found in the manufacture of laminated timbers and tennis rackets.

A typical laminated timber is made using six 1×2 -in. pieces and gluing these on their 2-in. faces to form a 2×6 -in. piece. It is far more economical in this case to set the glue by heating along the glue line rather than by heating the entire mass of wood to set the glue. Both straight and curved laminated timbers are produced commercially by RF heating. In the case of the 2×6 timber made of six 1×2 's, electrodes approximately 5 in. wide are used, and their length depends upon the generator rating, which in turn depends upon the production rate desired. The length is usually proportioned to give heating times of around 30 sec. With straight timbers, only sufficient glue strength need be developed during the heating cycle to permit preliminary handling, and full strength is developed during normal delays before finishing operations. With curved timbers, additional glue strength must be developed while in the clamps in order to overcome the shearing forces on the glue line set up by the bent laminae. Production rates possible vary considerably and depend upon the type and size of the wood pieces, type of glue and glue area, and upon whether straight or curved timbers are being produced.

The manufacture of a tennis, badminton, or squash racket is a very interesting application of RF power. In order to produce the best rackets with freedom from warping and with proper resilience, strength and "feel," the manufacturers have found it best to make them from $1/8$ to $1/32$ -in. veneers of very highly selected hardwoods. In addition to the long veneers, each about 1 in. wide \times 65 in. long, a triangular-shaped throat, a crescent-shaped bow reinforcement and a handle plug or wedge must also be glued in. In the better rackets, several thin fiber pieces are added between the veneers for decorative effect and for reinforcement against splitting when the stringing holes are drilled. All these pieces are spread with glue and placed in pneumatic or mechanical clamps which form the veneers to the desired shape. In a 10-ply racket there are approximately 700 sq in. of glue area. Using flat electrodes, shaped to conform to the outlines of the racket, and heating along the glue lines, the racket can be set in about 30 sec with a 2-kw RF generator. The racket is complete and ready for further operations as soon as it is removed from the press.

DRYING

The present-day bottleneck in the woodworking industry appears to be in kiln-drying properly before fabrication. Unfortunately, RF heating is not economically sound for this application except in very specialized cases. It has been found by laboratory tests that approximately 3.25 lb of water can be evaporated per kilowatt-hour. Since approximately 2000 lb of

water must be removed from 1000 fbm of green lumber, 615 kw of RF power will be required to process 1000 fbm per hr.

For very specialized applications, as in the final drying of knife handles, tool handles, pipe blanks, golf-club heads, where conventional drying means may cause considerable rejects due to checking, splitting, or where accurate drying control is desired, RF heating has proved economically sound.

ASSEMBLY GLUING

One of the most interesting applications of RF heating in the woodworking industry is that of assembly gluing for the manufacture of items such as radio cabinets, clock cases, and sewing cabinets. With a little experience, the design of the jigs and clamps, location of the electrodes, and selection of the proper generator rating and frequency are relatively simple. The best way to describe this process is to study a particular case history. The radio-phonograph cabinet, shown in Fig. 7, is assembled in production by this process, using waterproof resin glues and RF heat. No nails or screws are used at all.

The first operation is the bonding and forming of the curved plywood front panel. Three plies of $1/16$ veneer, approximately 30 in. \times 22 in., are spread with glue and placed in a press. The press forms the veneers and a 5-kw 5-megacycle RF generator sets the glue in about 120 sec. This piece is then cut, making four cabinet front panels.

The next two operations are accomplished with one two-position 2-kw 15-megacycle generator with a transfer switch to switch the power alternately to two jigs. Both jobs are done with simple jigs using spring-loaded clamps for pressure. Since these operations require 30 and 20 sec, respectively, and the jig-loading times are 20 and 30 sec, the generator is in practically constant use. The generator is adjusted so that no change is required when changing jobs, and the two generator timers are set for the two required heating times. The first of these operations is to apply a $14 \times 6 \times 3/16$ flat plywood stiffener, a long stiffener piece on the top, a bottom edge piece and four small triangular blocks to the curved front panel. This is a combination of through-heating to glue the two pieces of plywood, and glue-line heating to fasten the other pieces and is done in 30 sec. The second of these operations is also a combination of through-heating and glue-line heating in which a thin strip and two triangular blocks are assembled to the record-player shelf.

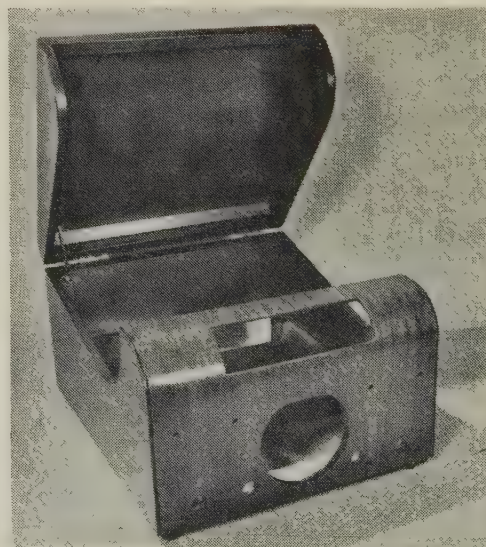


FIG. 7 RADIO-PHONOGRAPH CABINET MANUFACTURED AND ASSEMBLED BY RF HEATING

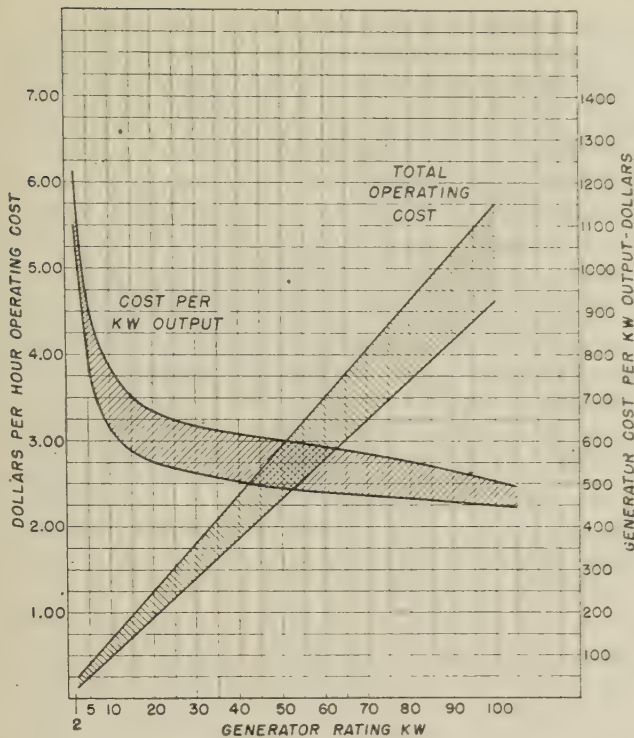


FIG. 8 CURVES SHOWING COST OF RF GENERATOR PER KILOWATT RATING AND TOTAL OPERATING COST

In the next operation the side panels are assembled. Two small rectangular blocks for the radio-chassis support, two rounded corner pieces to reinforce the curved front panel, the bottom edge piece, and a glue block to reinforce it are all glued to the side panel. Each of the small pieces is placed in a recess in the jig, the flat side panel is placed on top and located from the jig, and the whole assembly is placed in a press where an air cylinder exerts the gluing pressure. The electrodes are in the jig, and the press ram carries the RF leads to the jig. Two of these assemblies are heated together for 20 sec with a 5-kw 15-megacycle RF generator. Since the parts are placed in the jigs outside the press, several jigs are provided, and the generator is in constant use except for the press opening and closing times.

In the fifth operation the top of the cabinet is glued. Here the two sides, front strip, back, and glue block for the back are assembled in a jig, clamped and heated for 45 sec with a 2-kw 15-megacycle generator. Again, a transfer switch is used so that as one jig is being loaded and unloaded, the parts in the other are being heated and the RF power is on practically continuously.

In the sixth and final operation, the subassemblies for two cabinets, front, two sides, record-player shelf, back, and several glue blocks are loaded in a jig and heated for 110 sec with a 5-kw 15-megacycle generator. A transfer switch and two jigs are provided to permit practically continuous generator operation.

In all of these operations, the cost of the jigs and presses is remarkably low. Changes in cabinet design resulting in jig changes do not necessarily mean a costly retooling process or long interruptions in production. This process does not rely upon nails, screws, or clamps to hold the cabinet together for a long glue-setting period, and the absence of screws and nails results in a finer cabinet. In addition to the advantages mentioned, the use of RF power for glue-setting has also resulted in cabinets which are truly square and accurately aligned, floor-space requirements have been reduced by eliminating storage space, and, by better plant layout, labor costs have been reduced by using

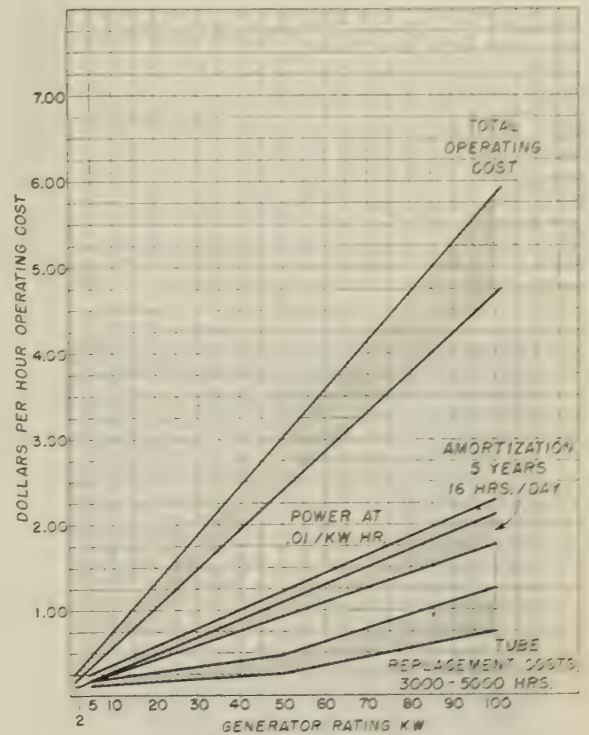


FIG. 9 CURVES SHOWING OPERATING COSTS OF RF GENERATORS

fewer and less skilled operators, fewer clamps, presses, and jigs are required, and stronger cabinets result, due to the use of water-resistant glues.

ECONOMICS OF RF HEATING

The cost of radio-frequency power per kilowatthour output, as in other electrical equipment, is not uniform but decreases gradually as the output rating increases. This is illustrated in Fig. 8. The total operating cost of the generators for various ratings is also shown in Fig. 8, and a breakdown of this total is shown in Fig. 9. The tube-replacement costs are based on a tube life of 3000-5000 hr, amortization on 5 years at 16 hr per day and power at \$0.01 per kw-hr. These curves are based on continuous full-power operation which is seldom achieved. When operating at reduced power or on intermittent processes, the power costs may be reduced and tube life extended. To the foregoing must be added the cost of associated equipment and labor costs. Labor costs are usually low since unskilled labor is used. The cost of associated equipment varies from a few dollars to many times the cost of the RF generator.

It is interesting to note that these curves are approximately 20 per cent lower than those published¹ in 1943, in spite of generally increasing prices. Mass production and use of RF power which is really just beginning, together with simplification and standardization of design and better tube performance should lower these costs still further.

THE RADIO-FREQUENCY GENERATOR

The purpose of the radio-frequency generator used in industrial heating is to take energy from a 60-cycle power source and convert this into energy at frequencies of many millions of cycles per second. The generator operates essentially as follows: It takes energy at 220, 440, or other commercial voltages, trans-

¹ "Heating Wood With Radio-Frequency Power," by J. P. Tamm, Trans. ASME, vol. 65, 1943, pp. 207-212.

forms it by means of a plate transformer to about 10,000 volts and converts this to a high-voltage direct-current supply through rectifier tubes. This high direct-current voltage is pulsed by an oscillator tube into a circuit resonant at the desired radio frequency. The resonant circuit or "tank," made up, generally, of capacitors and inductances, stores the energy and controls the frequency. The energy for heating is then drawn from the tank by the load matching or coupling network.

The design of the tank circuit and coupling network for efficient stable operation and for operation with the greatest variety of loads is the most important consideration in RF generator de-

mission line, power line, and work circuits may form an antenna and radiate energy into space. The radio-frequency generators used in the woodworking industry range in power from 2 kw to several hundred kilowatts and at the frequencies generally used, 3 to 30 megacycles per sec, only a very small portion of this power may produce sufficient radiation to cause interference to vital communications channels.

Under the Communications Act of 1934, the Federal Communications Commission was given control over all channels of interstate and international radio communications and also over all sources of radio-frequency power which might interfere with such communications. By 1945 the industrial radio-frequency power installed already exceeded the total power of all broadcast stations, and some communications interference was encountered, particularly from medical diathermy equipment. At the request of the Federal Communications Commission, a committee consisting of manufacturers, users, and consultants in the radio-frequency-heating field was set up to study this problem. A great deal of time was spent in making radiation tests on existing equipment and in formulating proposed rules.

The final rules of the Federal Communications Commission, which were made effective July 1, 1947, are as follows:

1 Several frequencies are assigned to radio-frequency heating, namely, 13.6 Mc, 27.3 Mc, 40.9 Mc, and some other much higher frequencies not now commercially important. Operation on these frequencies is permitted without a license.

2 When not operating on an assigned frequency, the installation

(a) Shall have a field-strength radiation of not more than 10 microvolts per meter³ at one mile from the source of radiation.

(b) When interference with communications channels is caused regardless of the amount of radiation, the radiation shall be reduced to such limits to prevent the interference.

Close tolerances are specified for operation on the assigned frequencies so that expensive automatic frequency control equipment must be incorporated in the generator to keep it within the frequency limits. This equipment is expensive and bulky and its use is limited economically to the larger RF generators of 50 and 100-kw rating. Moreover, most wood heating is done in the frequency of 3 to 10 megacycles and there is no assigned frequency in this range.

Therefore all installations of RF generators made after July 1, 1947, unless on an assigned frequency and frequency stabilized, must be properly shielded to prevent communications interference. The rules of the FCC further state that a certificate must be furnished by the manufacturer of the equipment or by a competent engineer stating that each unit of the installation, RF generator, press, etc., has been type-tested and can reasonably be expected to meet the radiation requirements. Installations made prior to July 1, 1947, are exempt from these rules except that if communications interference is found, the equipment will be shut down, the source of interference remedied, and a certificate be then issued stating that the equipment has been tested and found satisfactory.

These regulations may appear to be severe and discouraging to the prospective user of RF equipment. This is not so. The advent of these rules has been expected for some time by established manufacturers of RF equipment, and the better generators are built to minimize radiation. The radio industry has provided the engineering know-how on designing, placing, and shield-

³ The microvolt per meter is the unit of RF field strength. Approximately 25 microvolts per meter are required for satisfactory AM broadcast reception with a small receiver, 5 to 10 microvolts per meter for FM reception, and 1 microvolt per meter for reception with a highly sensitive communications receiver.



FIG. 10 WESTINGHOUSE 10-KW INDUSTRIAL RF GENERATOR DEVELOPED ESPECIALLY FOR EDGE-GLUING APPLICATIONS

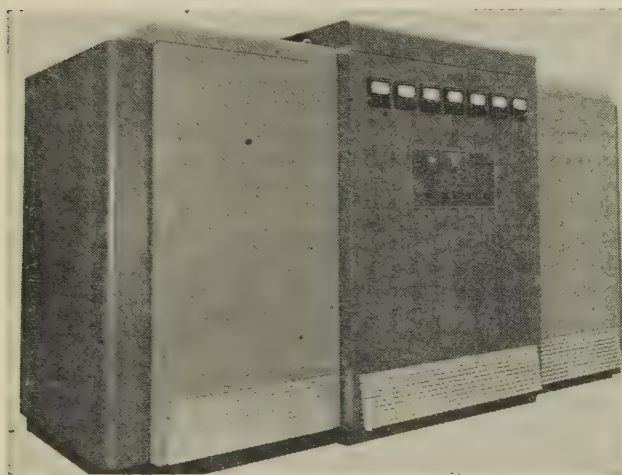


FIG. 11 WESTINGHOUSE 100-KW INDUSTRIAL RF GENERATOR

sign. A 10-kw and a 100-kw industrial RF generator are shown in Figs. 10 and 11, respectively.

FEDERAL COMMUNICATIONS COMMISSION REGULATIONS

The radio-frequency generator used for industrial-heating purposes is essentially the same as a broadcast or communications transmitter, and if connected to an antenna would radiate its generated energy into space. Although designed and intended to concentrate its energy into the workpiece to be heated, there is always the possibility that parts of the generator, RF trans-

ing the components outside the generator, which are the major causes of radiation. The shielding required to make the equipment safe, with a few minor refinements, also serves as the radiation shield. Field-strength tests made on equipments of 100-kw rating and higher have proved the soundness of this engineering knowledge, so that the average user of 10 and 20-kw generators need have no fear if his equipment is properly designed, installed, and operated.

CONCLUSION

The highly progressive and competitive woodworking industry offers perhaps the most fertile field for the application of RF heating. While RF heating is new, it has already made great strides in some woodworking applications and has barely scratched the surface on others. A great deal of co-operative research and development work must be done by the woodworker, glue chemist, and generator manufacturer before the potential benefits are all derived.

Radio-frequency-heating is not a panacea. It is just another method of heating, and 1 Btu of RF heat is exactly equal to 1 Btu of any other kind of heat, except that the radio-frequency Btu costs a lot more but can be directed and controlled to heat uniformly or only the areas necessary, resulting in a saving of heat required and thus gaining an advantage in over-all economy. As time goes on, we will have cheaper and more durable generators, with simple controls, and with greater ranges in frequency and other characteristics to cover a greater range of applications.

ACKNOWLEDGMENTS

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Results of Accelerated Tests and Long-Term Exposures on Glue Joints in Laminated Beams

By T. R. TRUAX¹ AND M. L. SELBO,² MADISON, WIS.

A rapid method developed at the U. S. Forest Products Laboratory for determining the durability of glue joints in laminated beams intended for exterior service is described. The test is based upon the principle that severe shrinking and swelling of laminated members produce stresses, which cause checking of wood, failure of glue joints, or both. Vacuum and pressure are employed in bringing about the desired conditions of shrinking and swelling. Results of the test are correlated with more extended laboratory exposures and with long-term soaking and weathering.

INTRODUCTION

THE development of new glues and gluing methods has been more or less paralleled with the development of new or modified test methods for their evaluation. Testing of glues dates back many decades and, undoubtedly, was practiced much earlier than existing records indicate. With each new development in the field of woodworking glues, there has arisen a need for new or modified test procedures which fully evaluate the new product; its properties, application, and usefulness.

Through the years some form of joint test has been the common and generally accepted method of testing most glues for wood joints and the adequacy of such joints for different uses. Many types of joint tests have been devised and used with variations in the form of specimens, the exposure or treatment of the joints before test, and the method of applying the load or stress to determine the strength and performance of the bond. The block-shear and plywood-shear tests, developed at the U. S. Forest Products Laboratory during the first world-war period, have been extensively used in determining the bonding strength of glues and the importance and limitations of gluing procedures.³ When the specimens were subjected to various exposures before test, the two methods, and especially the plywood-shear test, have also been valuable in determining the durability of glues and wood joints.

Production of water-resistant plywood during and following that war and its evolution into a type suitable for exterior use was preceded or accompanied by the development of accelerated test methods. The adequacy of plywood for exterior service was first established by subjecting unprotected panels to the variable conditions of outside exposure. The standards for its production were established, and the control of essential quality has been

maintained by the use of accelerated test methods carefully correlated to the performance of the panels in service. Likewise, in laminated construction there is in progress the evaluation of glues and gluing techniques under different service conditions and of testing methods that will assure the development of suitable standards and the maintenance of adequate glue bonds in service.

Laminated structural members for interior or limited exposures followed the development of water-resistant plywood in this country, but in Europe they have been in service since about the beginning of the century.⁴ Because of the form and size of material and essential differences in the details of construction of plywood and laminated members, however, the test methods and means of evaluating the two forms of construction must be varied. Furthermore, the methods in use for evaluation of the glue joints in interior laminated members are not adequate for judging the performance of joints under exterior exposure.

The necessity for test methods more discriminating in regard to durability of joints in heavy laminated construction than were provided by existing methods became apparent during the development of processes for laminating timbers for exterior use at the start of the recent war. Laminated oak timbers which gave initial high strength in the block-shear test soon developed delamination of glue joints when exposed to the weather.⁵ This led to the conclusion that test methods more nearly simulating service conditions would be necessary for evaluating the suitability of glue joints for exterior use.

Laminating for exterior use, like many other major developments, was prompted by an urgent need in time of war. The Navy needed immense numbers of ships and boats, the building of which soon brought on a shortage of available materials. White oak, the foremost shipbuilding species, seasons slowly, especially in large dimensions. Nominal 1-in. oak lumber, however, can be dried in a relatively short time. Therefore a process was developed whereby boat and ship timbers of the desired shapes and sizes and timbers for other exterior uses were laminated from kiln-dried lumber with intermediate-temperature-setting phenol and later with resorcinol resin glues.

In exterior use, timbers are subjected to wetting and drying which cause severe stresses and a tendency to check. In laminated timbers, these stresses are more or less concentrated at the glue lines. In addition, the laminations are usually relatively thick, in comparison with veneer used for plywood, and the requirements on glue-line performance are even more rigid than in exterior plywood. Hence accelerated tests that predict accurately the durability of laminated members under exterior exposure are of considerable importance.

The work of the U. S. Forest Products Laboratory on test procedures and test methods to be discussed, deals exclusively with

¹ Technologist, Forest Products Laboratory, Forest Service, U. S. Department of Agriculture.

² Chemical Engineer, Forest Products Laboratory, Forest Service, U. S. Department of Agriculture.

³ "The Gluing of Wood," by T. R. Truax, USDA Technical Bulletin No. 1500, June, 1929.

Contributed by the Wood Industries Division and presented at the Annual Meeting, Atlantic City, N. J., December 1-5, 1947, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 47-A-85.

⁴ "The Glued Laminated Wooden Arch," by T. R. C. Wilson, USDA Technical Bulletin No. 691, October, 1939.

⁵ "Laminating Lumber for Extreme Service Conditions," by C. D. Decker and A. C. Knauss, *Mechanical Engineering*, Vol. 64, 1944, pp. 763-773.

types of glues, such as phenol, melamine, and resorcinol resins which, when properly used, have been found durable in exterior service.

DEVELOPMENT OF ACCELERATED TEST METHODS

During service, timbers used for exterior purposes may be expected to undergo considerable change in moisture content. To furnish dependable test results, stresses similar to those which might occur in service therefore must be induced during the test and, furthermore, the possible leaching or solvent effect of moisture upon the glue must be simulated. The two accelerated tests, to be described in the following, were intended to simulate these conditions.

Soaking-Drying Cyclic Delamination Test (180-day test). Sections 2 in. long were cut from laminated beams 8×10 or 6×6 in. in cross section and exposed to a repeating cycle of soaking for 30 days, followed by drying in a room at 80 F and 30 per cent relative humidity for 30 days. The test included three complete cycles for a total test period of 180 days. Observations were made on the condition of the sections and, during the first week of drying in each cycle when the delamination was most in evidence, the total length of open glue joints on the sections was measured. This was expressed as a percentage of the total length of glue joints exposed, and referred to as the percentage delamination. A low percentage of delamination indicates durable joints, and a high percentage poor durability. Examples of good and poor durability as indicated by the 180-day cyclic test are shown in Figs. 1 and 2.

Vacuum-Pressure, Soaking-Drying Cyclic Delamination Test (21-day test). The chief disadvantage of the 180-day cyclic test was its extended duration and, consequently, attention was paid to development of a test method that would be less time-consuming. It was found that by application of vacuum and pressure the moisture content of 6×6 -in. beam sections 3 in. long could

be brought well above the fiber-saturation point and often as high as 80 per cent or more in a period of 24 hr. Since the checking in the wood and open glue joints usually were more pronounced during the first week of drying, it was assumed that the drying part of the cycle might also be shortened. Eventually, the following test procedure was evolved:

The test sections were placed in an autoclave or other type of pressure vessel; they were immersed in water at room temperature and weighted down to keep them submerged. A vacuum of 20 to 25 in. of mercury was then drawn and held for 2 hr. The vacuum was released and air pressure of 75 psi was applied for 2 hr. The vacuum-pressure cycle was repeated once, while the specimens remained immersed. The soaking was continued for an additional 16 hr at atmospheric pressure. The specimens were then dried in a room maintained at 80 F and 30 per cent relative humidity and provided with brisk circulation of air. The sections were placed at least 2 in. apart and with the end-grain surfaces parallel to the stream of air. The drying was continued for a total period of 6 days. The entire soaking-drying cycle was repeated twice for a total test period of 21 days.

The total length of open glue joints (delamination) on the end-grain surfaces of the sections was measured during the part of the drying period of the third and final cycle when the checking of the wood and open glue joints were most in evidence (usually during the third or fourth day of drying). To facilitate measurement of delamination, the end-grain surfaces of the sections were cut or sanded smooth prior to the start of the test. A magnifying glass was used as an aid in measuring delamination.

Examples of different degrees of durability, as indicated by the vacuum-pressure 21-day cyclic test, are illustrated in Figs. 3 and 4.

It is realized that variations in the 21-day test may be permissible without making the test less useful and further work is planned with the idea of shortening the test period. For example, two cycles requiring 14 days instead of the three cycles

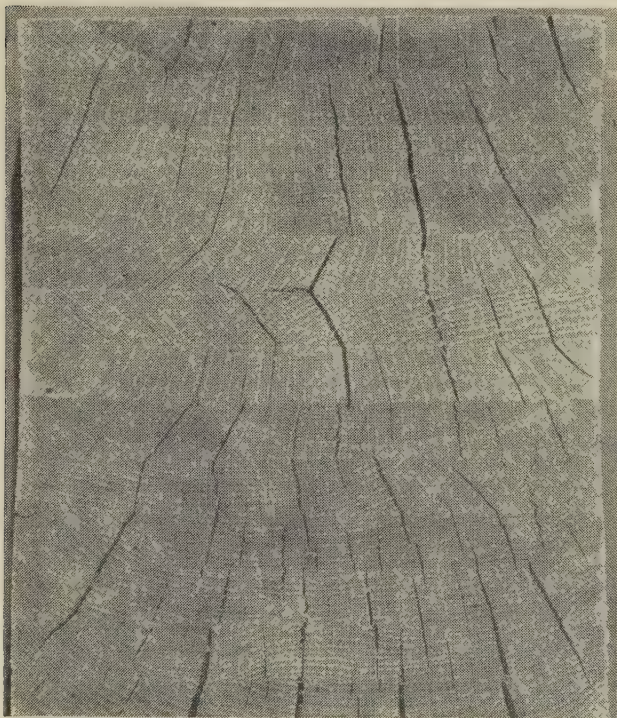


FIG. 1 CROSS SECTION OF WHITE-OAK BEAM, EXPOSED TO THREE CYCLES OF 30-DAY SOAKING AND 30-DAY DRYING TEST, INDICATES EXCELLENT DURABILITY OF GLUE JOINTS

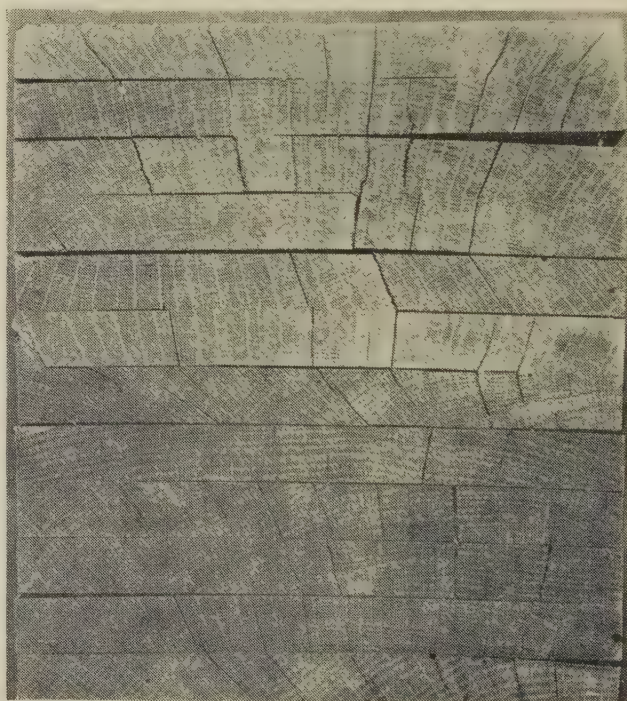


FIG. 2 CROSS SECTION OF WHITE-OAK BEAM, EXPOSED TO THREE CYCLES OF 30-DAY SOAKING AND 30-DAY DRYING TEST, INDICATES POOR DURABILITY OF GLUE JOINTS



FIG. 3 CROSS SECTION OF WHITE-OAK BEAM, EXPOSED TO THREE CYCLES OF VACUUM-PRESSURE SOAKING-DRYING TEST, INDICATES POOR DURABILITY OF GLUE JOINTS

TABLE 1 DELAMINATION IN WHITE-OAK BEAM SECTIONS AFTER EACH OF THREE VACUUM-PRESSURE, SOAKING-DRYING CYCLES

Cycle 1	Delamination, per cent	Cycle 2	Cycle 3
0.1	0.4	1.2	
.3	1.1	1.5	
.4	1.5	1.9	
.6	1.6	1.8	
.8	2.2	4.2	
1.2	2.1	2.5	
1.3	3.8	3.9	
1.4	2.9	4.4	
1.8	3.5	4.5	
2.6	5.9	6.5	
3.3	8.7	11.1	
3.9	5.5	6.4	
4.0	6.1	9.5	
4.6	9.5	11.9	
4.7	8.0	8.4	
7.5	15.5	16.4	
8.0	12.2	15.3	
9.2	11.9	15.2	
10.6	20.2	22.5	
12.6	17.5	20.0	
15.3	29.5	31.2	
16.5	20.5	23.5	
25.3	31.2	41.4	
27.8	33.8	36.2	
Average	6.8	10.6	12.6

requiring 21 days might be sufficient. Some available data, however, indicate that three cycles are preferable to two.

In Table 1 the amount of delamination developed in each of 24 white-oak beam sections after one, two, and three cycles in the accelerated vacuum-pressure, soaking-drying test is shown. The average delamination value for all sections after various cycles is shown graphically in Fig. 5. In general, the amount of delamination increased with each of the three cycles, but the average rate of increase was greatest in the first cycle and less in each of the two succeeding cycles, with an indication that the amount of further delamination beyond the three cycles would be small, provided the test was continued further.

Usually, after a few cycles of severe soaking and drying, the internal stresses are largely relieved, either by failure of the glue joints or by checking of the wood, and the amount of glue-joint delamination brought about by further exposure is relatively

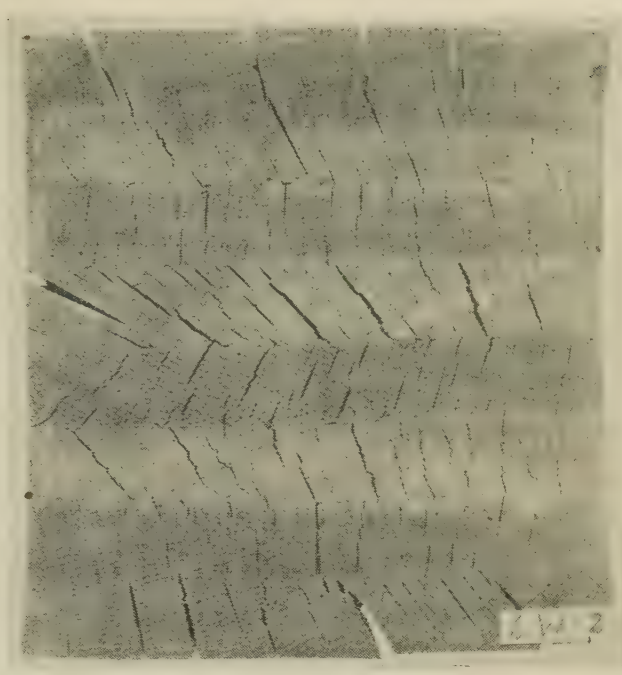


FIG. 4 CROSS SECTION OF WHITE-OAK BEAM, EXPOSED TO THREE CYCLES OF VACUUM-PRESSURE SOAKING-DRYING TEST, INDICATES EXCELLENT DURABILITY OF GLUE JOINTS

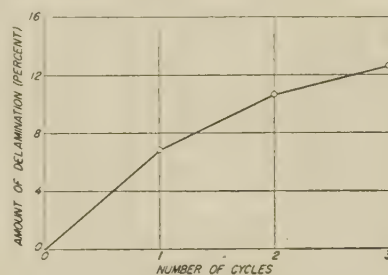


FIG. 5 AVERAGE AMOUNT OF DELAMINATION IN 24 WHITE-OAK BEAM SECTIONS AFTER 1, 2, AND 3 CYCLES IN VACUUM-PRESSURE SOAKING-DRYING TEST

small. On the other hand, if the test is carried through only two cycles where the rate of increase in delamination is still appreciable, there is less assurance that the results are true indications of the durability of the glue joints than when three cycles have been completed, and the delamination has more nearly reached its maximum.

LONG-TERM EXPOSURES

To be of value, any test method for laminated members intended for exterior service should produce results indicative of durability in such service. Long-term weather exposures and continuous soaking tests of laminated beams were therefore started early in the work as a control for other tests.

Weathering Tests. For the weathering tests beams were made 6×6 or 8×10 in. in cross section and about 4 ft in length. To one end of each beam a protective coating was applied, after which the beams were placed on racks and subjected to outdoor exposure with the unpainted ends directed south. At intervals of a year or less the unpainted end of each beam was inspected for delamination. At the time of the inspections, block-shear tests were usually made on sections cut from the painted ends of the beams.

Soaking Tests. Laminated beams 8×10 in. in cross section

and about 4 ft in length with a protective coating on one end were immersed in salt water (4 per cent) for periods up to $3\frac{1}{2}$ years. They were examined for delamination at regular intervals, and sections were also cut from them for block-shear tests. These sections were conditioned to about 12 per cent moisture content before the block-shear specimens were cut and tested.

CORRELATION OF RESULTS FROM VARIOUS TESTS

The ultimate aim of the various tests was to determine the durability of the glue bonds under severe exposure conditions and the test method that might be the most dependable for predicting durability, as well as being easily and quickly carried out. For this purpose a comparison of results from various test methods became necessary.

During the progress of the laminating studies, beams were made up from time to time with different glues and under different curing conditions so that they represented considerable variation in glue-bond quality. The beams were subjected to one or more of the four test procedures previously described, and usually two of the testing methods were applied to the same beams. In other cases duplicate beams, made at different times with the same glue and gluing conditions but tested by different methods, afforded a basis for comparison of testing procedures.

Comparison of Continuous Soaking and Weathering Exposures. Results of continuous soaking for periods up to $3\frac{1}{2}$ years of 60 some beams, Fig. 6, and weathering tests on a considerably larger number indicated that exposure to weathering produces delamination much more rapidly than continuous soaking. Only a few specimens among the beams exposed to soaking developed small amounts of open glue joints during the $3\frac{1}{2}$ -year period; whereas beams similarly made and exposed to the weather in many cases developed large amounts of open glue joints. Shear blocks cut from beams during soaking and reconditioned to about 12 per cent moisture content before testing, in general, showed no appreciable weakening, indicating that prolonged continuous soaking had very little deteriorating effect on the glue bonds. When beams which had been soaked for $3\frac{1}{2}$ years were dried, considerable delamination developed in some of them, but in others the glue joints remained intact even after severe drying. The beam shown in Fig. 7 had developed considerable delamination in practically every joint after 4 years of exposure to the weather; whereas a beam similarly glued and cured but exposed to continuous soaking was practically free from joint failure after $3\frac{1}{2}$ years of exposure. In the beams exposed to weathering, the glue joints that showed little or no opening during the first year usually remained in about the same condition throughout the several years of exposure.

Comparison of Results From 180-Day Cyclic Test and 21-Day Vacuum-Pressure Cyclic Test. In so far as the time element is concerned, the 21-day cyclic test is a great improvement over the 180-day test. The means of testing durability, by inducing stresses which would result in checking or delamination or both, was similar for both tests. The possible leaching or solvent effect might of course be greater for the longer soaking periods of the 180-day tests. In the continuous soaking tests of beams and also in previous work with plywood,⁶ it was established, however, that the effect of soaking on the types of glues that are durable under severe exposure is not appreciable. For this reason it was considered that the shortening of the soaking period would probably not affect the results. In Table 2 delamination values are shown for two different groups of test sections from beams similarly

TABLE 2 DELAMINATION VALUES IN TEST SECTIONS TAKEN FROM LAMINATED BEAMS SIMILARLY GLUED AND CURED, BUT SUBJECTED TO DIFFERENT CYCLIC TESTS

Delamination developed in—	
180-day test, per cent	21-day test, per cent
0.0 ^a	1.3
.0	2.1
.0	2.2
.0	2.6
.0	3.2
.0 ^b	2.3 ^b
.1	3.5
.6	5.5
.6	1.0
.8	1.7
.9	5.8
.6 ^b	3.5 ^b
1.2	2.4
1.9	4.3
3.9	2.2
4.2	1.8
4.4	2.5
3.1 ^b	2.6 ^b
5.5	4.1
6.5	4.1
11.1	6.1
11.9	6.0
14.8	16.5
10.0 ^b	7.4 ^b
15.2	8.4
15.8	20.3
16.4	10.2
22.5	23.5
31.2	20.0
20.2 ^b	16.5 ^b

^a Reading down the left column the delamination values for the 180-day test are arranged in ascending order, and in the right column delamination values are shown for sections similarly glued and cured but subjected to the 21 day test.

^b Average of 5 preceding values.

⁶ "Summary of Information on the Durability of Woodworking Glues," by F. F. Wangaard, Forest Products Laboratory Report No. 1530 (revised 1946).

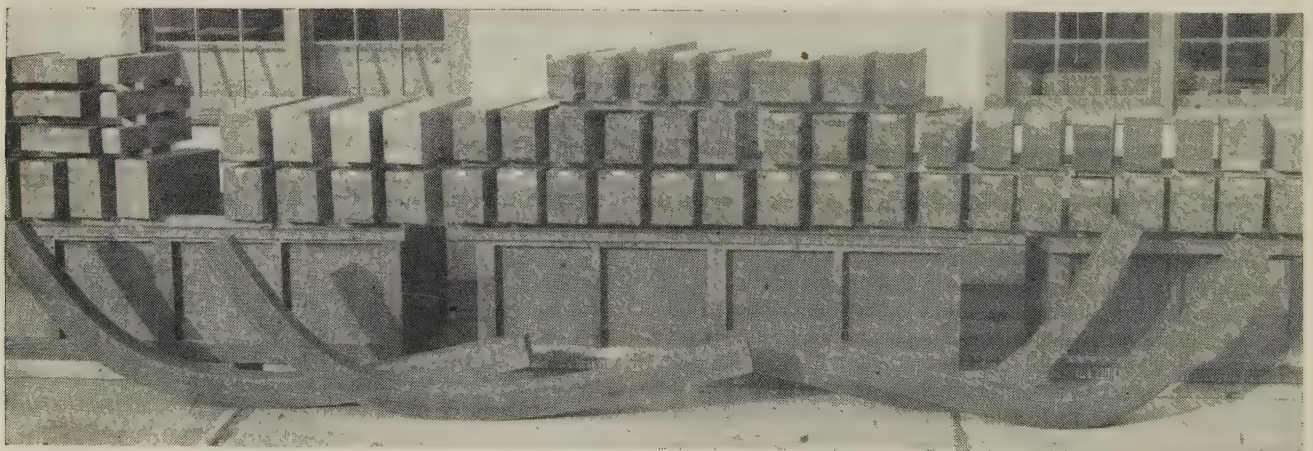


FIG. 6 LAMINATED BEAMS SOAKED FOR $3\frac{1}{2}$ YEARS IN SALT WATER OF 4 PER CENT



FIG. 7 LAMINATED WHITE-OAK BEAM EXPOSED TO WEATHER FOR ABOUT 4 YEARS SHOWS A HIGH PERCENTAGE OF OPEN GLUE JOINTS

glued and cured, one group having been subjected to the 21-day test and the other group to the 180-day test.

There is no very close agreement between the results from individual sections of each pair, but the general trend indicated by the two tests is similar. If averages for groups of five sections are considered rather than results for individual sections, the similarity becomes more obvious. The results show that where the delamination is small the test values from the 21-day test are slightly higher than those from the 180-day test. With increased magnitude of delamination, however, the reverse trend is indicated. It is possible that with highly undercured glues, the longer leaching period of the 180-day test may be more damaging to the joints; on the other hand, the differences in results are somewhat inconsistent and are probably not significant.

Comparison of Results of Weathering Test and Accelerated Vacuum-Pressure, Soaking-Drying Test. Since exposure to weathering causes deterioration of imperfect glue bonds (made with phenol, melamine, or resorcinol resins), it was used as a basis of comparison for the accelerated vacuum-pressure, soaking-drying cycles. It is recognized that weather conditions vary in different localities and result in a more severe exposure in one part of the country than in other parts. In the Madison area, where these tests were conducted, however, the climatic changes throughout the year are rather drastic, and the stresses induced in unprotected wood products due to temperature and moisture changes are quite severe. Consequently, it is believed that the glues and curing conditions which produced bonds that performed satisfactorily in unprotected outdoor exposure throughout the yearly cycle would be adequate for most exterior services; whereas appreciable delamination would indicate inadequate bonds. However, no attempt has been made to determine the relation of the severity of weathering period to the cyclic test in deterioration or delamination of glue joints.

In Table 3 a comparison is made between results of the 21-day cyclic test and weather exposure on white-oak beams. Percentage delamination (length of open glue joints) is shown in parallel

TABLE 3 AMOUNT OF DELAMINATION DEVELOPED IN WHITE-OAK BEAM SECTIONS SUBJECTED TO ACCELERATED VACUUM-PRESSURE CYCLIC TEST (21-DAY TEST), AND IN BEAMS EXPOSED TO WEATHER FOR APPROXIMATELY 1 YEAR

Delamination—		Delamination—	
In vacuum-pressure cycle, per cent	Under weather exposure, per cent	In vacuum-pressure cycle, per cent	Under weather exposure, per cent
0.5	3.1	10.2	28.0
0.6	4.1	11.9	12.5
0.9	4.4	12.0	28.0
0.9	9.6	14.8	42.8
1.1	7.0	16.0	54.0
1.2	5.9	16.5	40.0
1.3	11.4	17.8	37.8
1.4	9.9	19.2	27.0
1.7	10.9	22.1	55.4
2.0	14.0	25.5	46.6
2.2	11.7	29.4	44.0
2.2	13.0	48.0	100.0
2.3	18.7	76.0	95.0
2.4	5.7	78.0	96.0
2.7	6.7	92.0	95.0
2.9	5.9	100.0	100.0
2.9	13.3		
3.3	6.0		
3.6	7.2		
4.1	6.0		
4.1	20.5		
4.2	25.1		
4.3	4.0		
4.4	7.0		
4.4	23.9		
5.9	18.1		
6.0	27.0		
6.1	11.2		
8.2	13.5		
9.5	32.6		

columns for beam sections subjected to cyclic test, and the beams exposed to the weather for approximately 1 year. Similar comparative tests were made, using the 21-day cyclic and the weathering tests on groups of hard maple, southern yellow pine, and Douglas-fir beams. In most cases the specimen for the cyclic test was cut from the same beam that was exposed to the weather, and in every case the cyclic section represented the same species, glue, and curing condition that were used with the weather-exposure beam. In determining the amount of delamination in the cyclic test, both end-grain surfaces of the specimen were

TABLE 4 AVERAGE AMOUNT OF DELAMINATION BY GROUPS OF BEAMS IN VACUUM-PRESSURE SOAKING-DRYING CYCLE AND WEATHERING TESTS

Grouping by amount of delami- nation in cyclic test, per cent	White oak Delamination Per cent			Hard maple Delamination Per cent			Southern pine and Douglas-fir Delamination Per cent		
	No. of beams	Cycle	Weather- ing	No. of beams	Cycle	Weather- ing	No. of beams	Cycle	Weather- ing
0 to 1.0	4	0.8	5.3	3	0.8	4.5	28	0.2	1.0
1.1 to 2.0	6	1.5	9.8	7	1.7	6.8	7	1.8	2.2
2.1 to 5.0	15	3.3	11.6	8	2.3	12.7	9	3.0	4.0
5.1 to 10.0	5	7.1	20.5	2	6.6	18.8	5	7.0	10.7
10.1 to 20.0	8	14.8	33.8	4	11.7	33.6	4	12.6	13.0
20.1 to 30.0	3	25.7	48.7	0	2	24.8	27.6
Over 30.0	5	78.8	97.2	0	1	88.9	99.6

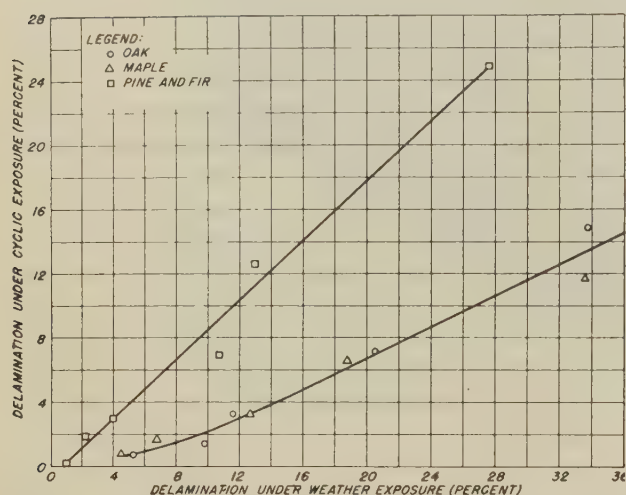


FIG. 8 COMPARISON OF DELAMINATION VALUES OBTAINED WITH VACUUM-PRESSURE SOAKING-DRYING CYCLE AND EXPOSURE TO WEATHER

examined and the condition of the glue joints recorded; whereas, in the weathering test, the amount of delamination in the glue joints was based only upon the one end surface of the beam facing south. This may have been an important factor in determining the amount of delamination recorded in the two tests and may have affected to some extent the consistency of results.

The results on white-oak beams, Table 3, show considerable divergence between individual specimens, and the same was true for the other species. However, the general trend of increasing delamination in the cyclic test with increased delamination under exposure to the weather was exhibited in all the tests. This trend becomes even more apparent when the averages of groups of specimens are taken. Such averages of groups for the different species (southern yellow pine and Douglas fir grouped together), shown in Table 4 and presented graphically in Fig. 8, indicate the general relationship between the two types of test. The pattern of the data suggests the probability of a somewhat different relationship for the hardwoods and softwoods, but the indications are not yet sufficiently definite to warrant positive conclusions.

The failure of glued joints is frequently the result of improper gluing technique, and, in the case of thermosetting glues to be used for exterior purposes, improper cure has often been found to be an important cause of failure. The vacuum-pressure soaking-drying cyclic test has been found convenient to determine the required curing temperature for different glues. Its use in determining the adequacy of joints in laminated white-oak beams, glued with two different intermediate-temperature-setting phenols is shown in Fig. 9, where percentage of delamination is plotted against the curing temperature.

The plotted values were obtained by applying the cyclic test to two 3-in. sections from each beam, while the remainder was

exposed to the weather. Hence each point on the graphs is based upon results from two specimens. From the graphs it may be noted that the amount of delamination is closely related to temperature of cure and was found to correlate well with the amount of delamination that developed on the parts of the beams exposed to weathering.

Results of block-shear tests on the same white-oak beams are also shown in Fig. 9. There was no significant difference in shear strength for the various curing temperatures. The wood-failure values, however, decreased generally with lower curing temperatures and appeared to disclose incomplete cure and poor durability better than the strength results.

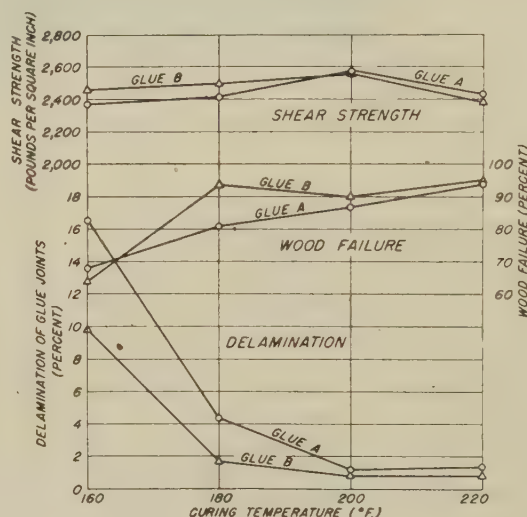


FIG. 9 RESULTS OF VACUUM-PRESSURE SOAKING-DRYING CYCLIC TESTS AND BLOCK-SHEAR TESTS ON WHITE-OAK BEAMS GLUED WITH TWO DIFFERENT ADHESIVES CURED AT VARIOUS TEMPERATURES

The appearance of beam sections and of the beams after the 21-day cyclic and weathering tests, respectively, are shown in Figs. 3, 4, 7, 10, 11, and 12. Figs. 3 and 4 show white-oak beam sections subjected to the vacuum-pressure cycle test, in which poor and good joint durability are indicated. Figs. 7 and 10 show two similar beams after exposure to the weather for 4 years with comparable results. Figs. 11 and 12 show six white-oak beams that represent a wide range in joint performance and durability when exposed to the weather, with the percentages of delamination obtained in the 21-day cyclic and weathering tests given in the captions.

Extensive block-shear tests were made on many of the beams immediately following their manufacture and at various stages during exposure in the other forms of test. Most of the results of such shear tests are not here presented because they were not found to indicate accurately joint performance under severe exposures. While the block-shear test is of value for the control of original joint quality, no close correlation exists between original joint strength and later delamination in weathering tests.

FIG. 10 WHITE-OAK BEAM EXPOSED TO WEATHER FOR ABOUT 4 YEARS SHOWS EXCELLENT DURABILITY OF GLUE JOINTS

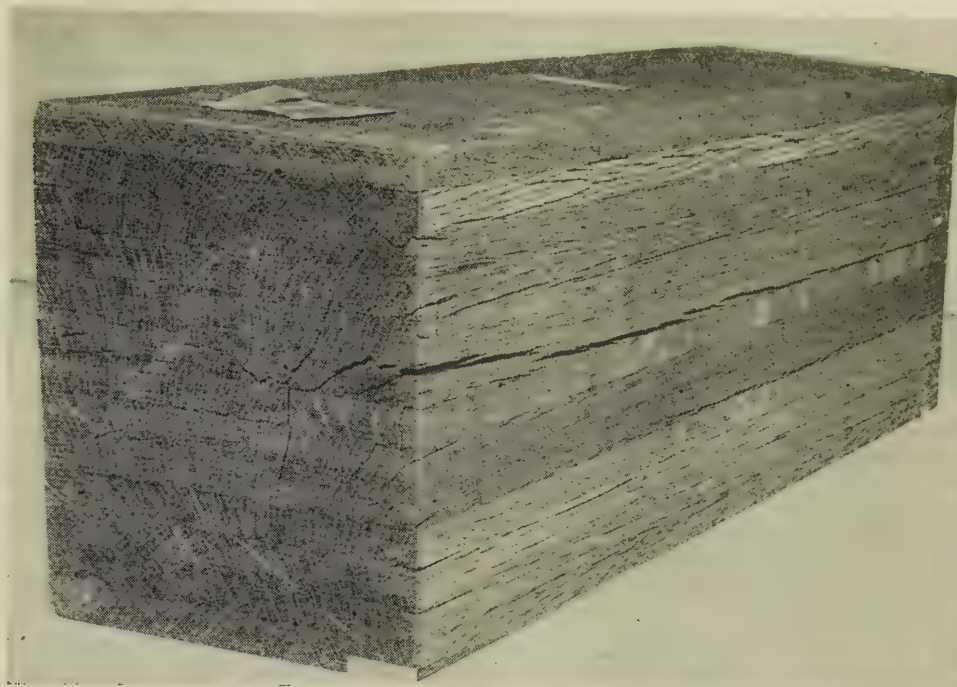


FIG. 11 LAMINATED WHITE-OAK BEAMS EXPOSED TO WEATHER FOR ABOUT 2 YEARS, FROM LEFT TO RIGHT, SHOWED PERCENTAGES OF DELAMINATION OF 100, 95, AND 9.3, RESPECTIVELY (Delamination values obtained in 21-day cyclic test on sections cut from these beams were 92, 76, and 3.6, respectively.)

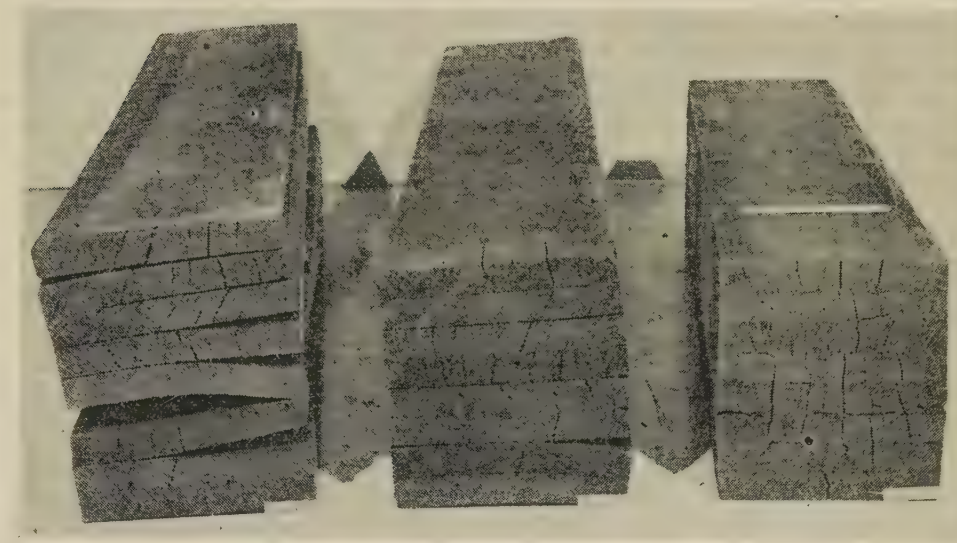
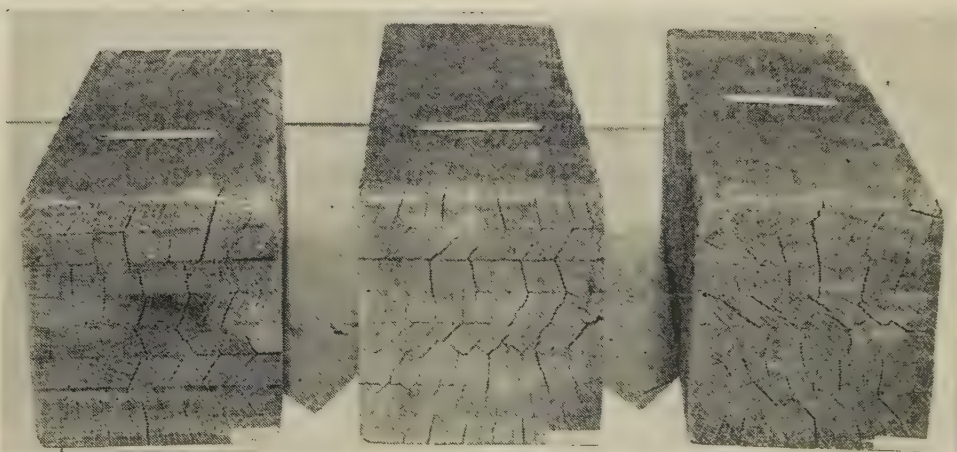


FIG. 12 LAMINATED WHITE-OAK BEAMS, GLUED WITH THREE DIFFERENT GLUES AND EXPOSED TO WEATHER FOR ABOUT 3 YEARS, FROM LEFT TO RIGHT, SHOWED PERCENTAGES OF DELAMINATION OF 15.7, 11.8, AND 8.5, RESPECTIVELY

(Delamination values obtained in 21-day cyclic test on sections cut from these beams were 6.1, 3.6, and 1.5, respectively.)



Neither is block-shear strength after exposure as reliable an indication of joint durability as the percentage of delamination in accelerated cyclic tests. The estimated wood failures in the block-shear test have been found to be a better indication of durability under severe exposure than the shear-strength values when glues of known durability are used. Estimated wood failures in the dry block-shear test, however, are of no value in distinguishing between durable and nondurable glues.

CONCLUSIONS

In conclusion, it appears that the vacuum-pressure, soaking-drying cyclic test gives a reliable indication of glue-joint durability in laminated beams under severe exposure; that it is fairly comparable in severity and results obtained with a longer soaking-drying cyclic test; that wetting-drying cyclic tests are more severe than continuous soaking and more nearly comparable to weathering as an indication of joint performance in heavy timbers; and that the block-shear test, while of value in judging original joint strength, does not indicate joint durability under weather exposures as accurately as percentage of delamination in accelerated cyclic tests, but that details of the vacuum-pressure, soaking-drying cyclic test are not yet fully known, especially as related to different species.

Discussion

C. F. MACLAGAN.⁷ When we first learned of the development of this "short-term" testing method, through visits to the authors' laboratory, we became considerably interested in it since the "quality of the bond" is measured without employing elevated temperature which in many instances further cures an incompletely cured glue line rather than determines the actual glue-line quality. Thus the necessary equipment was installed and many tests of this order have been carried out. Comparative tests between the 21-day vacuum-pressure procedure and 1-year unprotected outdoor exposure on a rack on our laboratory roof in Bainbridge, N. Y., reveal a fairly close correlation. This difference in our results over the results obtained at Madison may very well be due to the difference in climatic conditions.

What has been the experience in the effect of the thickness of the individual laminates making up the laminated beam section for this test? Does not the test become more stringent as the thickness of each laminate is increased?

What have been the experiences with regard to the results obtained on different species? For example, has it not been found that an adhesive exhibiting considerable delamination on white oak may exhibit no delamination on Douglas fir?

⁷ Casein Company of America, Research and Development Laboratories, Bainbridge, N. Y.

What are the authors' views regarding the adaptation of this testing method for such constructions as plywood to stud sections, etc., as employed in prefabricated house building?

Referring to methods of shortening the time factor for this type of cycle testing, has any consideration been given the use of 3 days' drying at 100 F rather than 6 days at 80 F. It is the writer's belief that the use of 100 F would not be instrumental in promoting further curing.

AUTHORS' CLOSURE

In a study not reported on here laminated beams were prepared from laminations varying over a considerable range both in width and thickness. The beams were fabricated according to the best-known technique with a durable glue and, when subjected to cyclic tests similar to those described here, no appreciable amount of delamination developed in any of them.

The stresses developed in a laminated section depend, among other factors, on the dimensions of each lamination. However, variations in stresses for different constructions occur not only during accelerated exposures, but also in service. Hence it is important that a test indicate the performance of the glue bonds, irrespective of the thickness of laminations or other construction details and the ability of the glue bonds to withstand the stresses developed in the particular construction under service conditions.

In regard to results obtained with different species, it is true that joints in certain species fail while in others, although made with the same adhesive and under the same conditions, no failure develops.* Offhand this might seem as if the test would be applicable to the one species, but not to the other. However, long-term weather exposures have confirmed that glues and gluing conditions that produce adequate bonds in one species may be unsatisfactory for another, so in reality the test gives a true indication of the glue-bond requirements for each species.

The purpose of the test is to bring about rapid failure where under ordinary exposure conditions failure would eventually develop after various periods of service, and it is entirely possible that it might be adapted to testing of other than laminated construction. In fact, the authors have used a modification of this test to evaluate the glue bonds in boat-frame assembly joints (frames joined with plywood gussets). It must be kept in mind, however, that in its present form the test is necessarily severe and that for constructions not intended for exterior service modifications would be required.

Shortening of both the soaking and drying periods of the cycle has been tried and appears to be feasible provided the drying conditions are favorable. A drying temperature somewhat higher than 80 F has also been used and is especially beneficial where it is difficult to maintain a low humidity.

Studies in Boundary Lubrication—II

Influence of Adsorbed Moisture Films on Coefficient of Static Friction Between Lubricated Surfaces

By W. E. CAMPBELL¹ AND E. A. THURBER,¹ MURRAY HILL, N. J.

Extraordinarily high values of the coefficient of static friction between steel surfaces lubricated with straight chain normal hydrocarbons (heptane to decane) are shown to be due to a film of adsorbed moisture at the solid-liquid interface. Friction measurements on a highly refined paraffin-base oil applied to surfaces in equilibrium with a dry and a moist atmosphere, confirm this idea, the coefficient of friction being 0.35 for the dry and 0.6 for the moist lubricated surfaces. Friction measurements are recorded for four fundamentally different lubricant types on steel, brass, and glass in equilibrium with dry air and air of 75 per cent humidity, respectively. The friction is in all cases raised from 40 to 70 per cent in the high humidity, the effects being particularly pronounced on glass. The results of the tests provide an explanation for many of the discrepancies among oiliness results in the literature obtained by the static-friction method, and indicate that the humidity of the atmosphere plays a significant part in the mechanism of boundary lubrication.

INTRODUCTION

THE coefficient of static friction between lubricated surfaces under boundary conditions has been shown by Rayleigh (1)² to increase when water is added to the lubricant. Hardy (2) later made a quantitative study of water solutions of certain organic liquids over a wide concentration range and found that the friction was raised by the presence of water. More recently Rhodes and Allen (3) found that water vapor in the air caused a rise in static friction, and in the first paper of this series (4) similar effects were noted. At no time, however, has a thorough investigation of the effect of adsorbed moisture films on the coefficient of static friction been made for commonly employed lubricant types. It is the object of this paper to record the results of such an investigation.

a desiccator through which dry air could be passed while a run was in progress. The capacity of the desiccator was 10 liters and the rate of air flow, in general, of the order of 100 liters per hr. Repeat measurements were made by withdrawing a stop 0.01 in. from the slider. Successive measurements could thus be made without admitting atmospheric air to the surfaces. The slider consisted of three 1-in. chromium-steel balls clamped at the apexes of a triangle between two steel plates. The plate was of hardened tool steel having an optically plane surface.

The procedure in making a run was to clean the surfaces of the balls and of the plate by an abrasive method, the test for cleanliness being the ability of water to wet surfaces uniformly in a very thin film (4, 5). After rinsing with redistilled alcohol and drying in a current of clean dry air, the slider was placed upon the plate, which in turn was placed upon a leveling platform mounted inside the desiccator. Dry air was then passed at a rapid rate for 15 min, and two or three preliminary clean-friction determinations made in order to check the cleanliness of the surfaces. If these measurements indicated that the surfaces were clean ($\mu = 0.78 \pm 0.04$), the lid of the desiccator was raised, the slider removed, and the balls shifted to expose a fresh surface of contact. A sufficient amount of the liquid to be tested to form a pool completely surrounding the balls of the slider when in position was added to the plate and the slider placed in position. The desiccator was closed and dry air was passed again from 10 to 20 min before starting friction readings. The rate of flow of air through the chamber was then cut down to 60 liters per hr in order to reduce the rate of evaporation of the lubricant. The paraffins used in these studies were obtained from the U. S. Bureau of Standards and were of very high purity. They were redistilled over sodium and kept sealed in pyrex tubes until ready for use.

MOISTURE EFFECTS FOR PURE HYDROCARBONS ON STEEL

The early results on octane, nonane, and decane gave initial

TABLE 1 RESULTS OF TESTS ON OCTANE AND DECANE

Lubricant		μ	0.76	0.75	0.70	0.64 ^a	<0.3 ^b	<0.4 ^b	0.50	0.51	0.53
Octane	Coefficient of static friction.	μ	0.76	0.75	0.70	0.64 ^a	<0.3 ^b	<0.4 ^b	0.50	0.51	0.53
	Time, min.	t	2	12	16	21	25	32	36	42	50
Decane	Coefficient of static friction.	μ	0.88	0.74	0.64	0.65	0.58 ^c				
	Time, min.	t	10	20	30	40	50				

^a No visible traces of lubricant left on plate.

^b Slider slipped at initial load applied.

^c Small pool of lubricant under each ball.

DESCRIPTION OF METHOD

The method used was essentially that of Hardy and has been described in detail in the earlier paper (4). It consisted in determining the load necessary to make a three-legged slider slip on a polished horizontal surface. The experiments were carried out in

coefficients of friction varying between 0.7 and 0.8, indicating that, under the conditions of the experiment, these liquids were nonlubricants on steel. These results were quite unexpected in the light of Hardy's measurements, which gave values varying between 0.35 and 0.45 for similar substances. In later runs on octane and decane, readings were continued until the bulk of the lubricant had evaporated. It was noticed that the readings became lower as the pool of lubricant became discontinuous due to evaporation from between the balls, and that the downward trend was accelerated as the last visible traces of lubricant vanished from the surface. Results indicating this effect are given in Table 1.

In still later work, measurements were continued for some time

¹ Bell Telephone Laboratories, Inc.

² Numbers in parentheses refer to the Bibliography at the end of the paper.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 47-A-18.

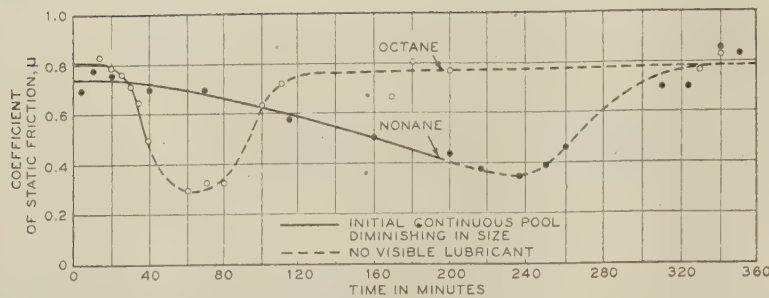


FIG. 1 COEFFICIENT OF STATIC FRICTION STEEL—HYDROCARBON—STEEL AS A FUNCTION OF TIME

TABLE 2 RESULTS OF TESTS ON OCTANE AND NONANE

Lubricant	μ	μ	μ	μ	μ	μ	μ	μ	μ
Octane	0.83	0.79	0.76	0.73	0.65 ^a	<0.5 ^b	0.30	0.33	0.33
	15	20	25	30	35	40	60	70	80
	0.63	0.72	0.68	0.82	0.80	0.78	0.77	0.82	0.83
	100	110	170	180	190	200	330	340	350
Nonane	0.70	0.78	0.77	0.72	0.70	0.70	0.59	0.51 ^a	0.45
	5	10	20	30	40	70	115	160	200
	0.35	0.40	0.47	0.74	0.74	0.87	0.82		
	235	250	260	320	350	340	350		

^a No visible traces of lubricant left on plate.

^b Slider slipped at initial load applied.

after all traces of visible lubricant had disappeared. The results given in Table 2 and plotted in Fig. 1 are typical of the behavior observed in these runs. Similar results were obtained for very pure butyl alcohol, except that the initial coefficient was not quite so high, being of the order of 0.6. It will be noticed that the results pass through a minimum and then trend upward, reaching a constant value equal to the clean value for steel on steel. Water run over the slider and plate at the end of the run remained in a continuous thin film, indicating freedom of the surfaces from adsorbed organic films.

These results may be explained in two ways: (a) on the assumption that evaporation of the hydrocarbon deposited a film of volatile impurity on the surface which is subsequently removed by evaporation. This seemed rather unlikely, since volatile organic impurities would be low-molecular-weight compounds and would not be expected to give friction values as low as those recorded. An alternative explanation (b) which seemed more in line with the facts observed, was that a film of moisture was trapped at the hydrocarbon-metal interface when the lid of the chamber was removed for placing the lubricant and slider in position, and was responsible for the high initial friction values recorded. The water evaporated after the bulk of the hydrocarbon had disappeared. The friction at the minimum point was that due either to a mixed hydrocarbon-water film or to a hydrocarbon film in equilibrium with the hydrocarbon vapors in the chamber. With further passage of air, the last traces of hydrocarbon were removed from the surfaces and the friction rose again to the clean value for steel on steel.

On this latter basis, a steady value of μ , much lower than the initial values previously recorded, should be obtained when a film is deposited on the surfaces from the hydrocarbon vapors alone. To test this point, the slider and plate were placed in the chamber after cleaning, and allowed to dry thoroughly, the slider being mounted on a pedestal out of contact with the plate. After $1/2$ -hr passage of dry air, the stream was diverted so that it bubbled through the hydrocarbon before entering the chamber. After thoroughly saturating the atmosphere in this manner, the slider was manipulated onto the plate without opening the chamber. Using this method, the following results were obtained: Heptane, $\mu = 0.57$; octane, $\mu = 0.48$; decane, $\mu = 0.44$, each result being the mean of 10 readings showing no trend of any kind. These results compare with values of $\mu = 0.43, 0.41, 0.39$, re-

spectively, obtained by Hardy (6) on steel surfaces for which $\mu = 0.74$.

MOISTURE EFFECTS FOR UNLUBRICATED SURFACES

The latter of the two explanations just given serves to account for the downward trend observed in earlier work (4), where the measurements were made in a dry atmosphere but the oil was placed on the surfaces before drying. Furthermore, in any series of dry-friction readings taken by the standard technique above, the first reading was invariably higher by a factor of about 10 per cent. It appeared that even, with clean surfaces, water adsorbed from the surrounding atmosphere before drying the

chamber was held between the balls and the plate, and could produce increased friction. To test this possibility further, the following experiment was carried out:

The slider and plate were dried in the desiccator out of contact with each other. The slider was then placed on the surface without opening the chamber and five readings were taken in the resulting very dry atmosphere. The readings were $\mu = 0.75, 0.77, 0.73, 0.80, 0.75$. The lid of the desiccator was now removed, the slider lifted out, and, after a few seconds' exposure to the room atmosphere (relative humidity 52 per cent), replaced on the plate. The lid was replaced and dry air passed for 15 min. Three readings using this procedure between readings gave $\mu = 1.05, 0.94, 1.16$. Three additional readings without reopening the chamber gave values of $\mu = 0.78, 0.73, 0.79$.

A similar experiment with a silver slider on a nickel plate gave even more striking results. Five readings with both surfaces dry in a dry atmosphere gave $\mu = 0.40, 0.37, 0.38, 0.37, 0.40$. Two readings taken by exposing the surfaces briefly to air and subsequent drying in contact gave $\mu = 0.84$ and 1.00.

Further readings taken without reopening the chamber gave $\mu = 0.38, 0.36, 0.39, 0.37, 0.37$. The chamber was now opened again, slider and plate breathed upon gently, and the slider replaced immediately after the visible moisture had evaporated. After from 30 to 50 min drying, readings were taken. The results for three measurements carried out in this manner were $\mu = 2.13, 1.57$, and 2.57. Upon opening the chamber and taking readings with the whole apparatus exposed to room humidity, a value of $\mu = 0.34$ was obtained as the mean of five readings and breathing upon the surfaces had no effect.

The nature of the slip was entirely different when the high readings were recorded, being very sharp and sudden as contrasted with relatively even movement for the lower values. The additional force required to produce slip, after exposure of the surfaces to moisture and subsequent drying, must be due to an invisible film of moisture adsorbed near the zone of metallic contact. Over the rest of the surface the film is absent. The area at which the water is held must be very small and the high friction cannot be due to ordinary capillary forces. Possibly chemical bonds have to be destroyed before slip occurs.

The situation for an adsorbed water film at an oil-metal interface is different in that the water film extends over both surfaces. The mechanism in this case will be discussed later. The clean-friction results provide a demonstration, however, of the enormous effects which can be produced by an invisible adsorbed film of water.

MOISTURE EFFECTS FOR WHITE OIL ON STEEL

In many of the earlier studies on reproducibility, highly refined white paraffin-base oils were used because they were considered to represent a comparatively simple and reproducible type of commercial product. Therefore the first studies of moisture effects were made upon such an oil which had been

percolated through Fuller's earth. This oil was then divided into two portions, one of which was shaken with freshly distilled water and allowed to stand in contact with water until ready for use. The other portion was dried by shaking with anhydrous sodium sulphate. Measurements of the static coefficient were then made for the following combinations:

- 1 Dry oil on dry surfaces.
- 2 Moist oil on dry surfaces.
- 3 Dry oil on moist surfaces.
- 4 Moist oil on moist surfaces.

In making dry measurements, the surfaces were dried out of contact as described earlier, the dry oil was added to the plate without opening the chamber, and the slider then manipulated into position. For the measurements on the moist surfaces, the slider and plate were allowed to come to equilibrium with degreased air at 50 per cent relative humidity before addition of oil. For measurements using dry oil, dry air was passed at 100 liters per hr during the course of the run; when moist oils were used, degreased air at 50 per cent relative humidity was passed. The oils were spread to cover the entire surface of the plate by means of a clean glass rod inserted through the air-exit hole. The graphited oil was the white mineral oil to which a commercial suspension of colloidal graphite in oil had been added to give a concentration of 0.02 per cent.

TABLE 3 EFFECT OF MOISTURE ON COEFFICIENT OF STATIC FRICTION BETWEEN STEEL SURFACES LUBRICATED BY A WHITE MINERAL OIL

Condition of oil	Condition of surfaces	μ	Average deviation from mean, per cent
Dry	Dry	0.33	2.3
Dry	Dry	0.36	3.4
Dry	Dry	0.35	4.9
Moist	Dry	0.37	4.5
Dry	Moist	0.61	2.7
Moist	Moist	0.62	2.1
Moist	Moist	0.60	6.4
Graphited	Moist	0.60	10.6
Graphited	Moist	0.54	7.0

In all the measurements, which are recorded in Table 3, the mean of 10 readings was taken after the friction had reached a steady value, since slight initial trends were sometimes noted. Although the reproducibility of a given set of 10 readings for the dry oil on dry surfaces is not appreciably better than that obtained in earlier work on undried oils and surfaces, considerably closer checks were obtained for the means of repeat determinations.³

The results confirm the effects observed for the hydrocarbons. It appears that the presence or absence of a moisture film on the metal surfaces before addition of oil is the important factor, because low results were obtained for a moist oil on a dry surface, while high results were obtained for a dry oil on a moist surface.

It is claimed that the graphite in a graphited oil adsorbs on the surfaces producing a "graphoid" surface of very low friction, and that such surfaces suffer correspondingly low wear. The results for the two measurements on a graphited oil indicate that no such film forms under the conditions of the experiment. The second group of measurements were made after rubbing the slider over an area of about 9 sq mm surrounding each ball for 10 min, allowing the surfaces to stand in contact for 24 hr, and repeating the agitation. Readings were then made on the portions of the surface which had been abraded in this manner. Though some lowering is evident, it is not sufficient to be caused by a graphite film,⁴ and could be accounted for by the presence of

³ Early measurements on a similar white paraffin-base oil had yielded values varying from 0.26 to 0.47.

⁴ Measurements of μ for the same slider against a graphite surface gave a value of 0.25.

polar constituents in the graphite suspension. It seems clear that if a graphoid film forms in a bearing under operating conditions, its formation must require considerable time and running-in.

MOISTURE EFFECTS FOR VARIOUS SOLID-LUBRICANT COMBINATIONS

Having established the mechanism of the friction rise produced by moisture films in the case of a type of oil known to be weakly adsorbed on steel, experiments were carried out to determine the nature of the effect for a series of typical oils on steel and brass. The same oils were then tested on glass surfaces, which were used extensively in Hardy's work, and which are of interest in mechanism studies because of their hydrophilic nature.

The static friction between the lubricated surfaces was determined for the dry oil on the dry surfaces by the procedure already described. For moist runs the surfaces were allowed to come to equilibrium with an atmosphere of 75 per cent humidity before addition of oil. The high-humidity measurements were carried out in a constant-humidity room at a temperature of 25 C. The surfaces were cleaned and dried by the standard procedure and were then allowed to come to equilibrium in the moist atmosphere for $\frac{1}{2}$ hr, during which time several clean values for μ were determined. It is of interest to note that these clean values are the same as the clean values in a dry atmosphere for each of the three solids. The brass slider was similar to the steel slider, 1-in. brass balls being clamped in a duplicate steel frame. The glass slider consisted of three clear glass marbles $\frac{5}{8}$ in. diam set in a steel frame. The brass plate was flat to 1×10^{-5} in. and was polished free of visible surface scratches. The glass plate was cut from a piece of plate glass.

The values obtained from these measurements are given in Table 4. They generally represent the mean of the first 10 values obtained for a given combination and set of conditions.

The results for repeat runs on the moist surfaces were not always very reproducible, particularly in the case of lard oil and oleic acid on glass. The lowest moist value, however, was always considerably higher than the corresponding dry value. Where more than one determination was made, the mean is recorded in Table 4.

TABLE 4 EFFECT OF MOISTURE ON COEFFICIENT OF STATIC FRICTION BETWEEN LUBRICATED METAL SURFACES

Lubricant	Steel		Brass		Glass	
	Dry	Moist	Dry	Moist	Dry	Moist
White mineral oil...	0.35	0.56	0.30	0.56	0.27	0.95
Spindle oil.....	0.27	0.46	0.27	..	0.16	0.70
Lard oil.....	0.11	0.33	0.11	..	0.16	0.85
Oleic acid.....	0.12	0.20	0.15	0.34	0.16	0.81
μ Steel-steel = 0.76						
μ Brass-brass = 0.86						
μ Glass-glass = 0.96						

It will be noticed that the friction-raising effect of moisture is present in every case, and that for steel and brass μ for the moist lubricated surfaces rates the lubricants in the same general order as μ for the dry lubricated surfaces. Adhesion tension work of Bartell (8) and preferential wetting measurements by Bulkley and Snyder (7), have shown that a glass surface is very hydrophilic and adsorbs water more strongly than fatty acids. As would be expected from these results, the friction-raising effect for glass is greater and more persistent than for steel and brass. The lower result for the spindle oil is rather surprising, but this oil showed unexpectedly low values in several other experiments, and the values for the dry runs showed a wider spread than any other oil tested. It would appear that this oil contains constituents giving it very high oiliness, but, unfortunately, it was not possible to obtain detailed information which could be used to explain its behavior.

It is worthy of note that there is no definite effect of the base metal in the dry measurements in Table 4, such as that which has been observed by Hardy and his co-workers (9, 10, 11).

These workers have also reported a decreasing trend of values of static friction for metal surfaces lubricated with polar compounds. The time for equilibrium to be established varies from 10 to 16 min and is called the "latent period." This is interpreted as representing the time for the lubricant molecules to rearrange themselves in a position of preferred orientation at the metal-liquid interface. Some evidence of such a downward trend was observed in making the dry measurements herein reported, in spite of the careful attempts to exclude moisture. However, the time to reach equilibrium was not sufficiently reproducible to be given a definite value, and was, in general, of the order of several hours. Moreover, although usually absent in measurements on nonpolar liquids such as white mineral oil, and somewhat more marked for substances such as fatty acids, the trend was present in many cases where it would be expected, from Hardy's work, to be nonexistent or at most very slight. In view of the trends obtained for the hydrocarbons, and those to be described for the moist measurements later, it is felt that the effects observed can be explained as being due to traces of moisture or to the formation of a corrosion layer at the metal-liquid interface.

REVERSIBILITY OF MOISTURE EFFECT

In order to determine to what extent changes in friction could be reversed by variation of the water in the atmosphere, several experiments were run on surfaces flooded with the white purified oil. It was found that changes in friction took place very slowly, in the absence of agitation, as was to be expected from the results for a moist oil on a dry plate. The results, plotted in Fig. 2, are the most convincing that were obtained. In this experiment air was bubbled through water and through the chamber

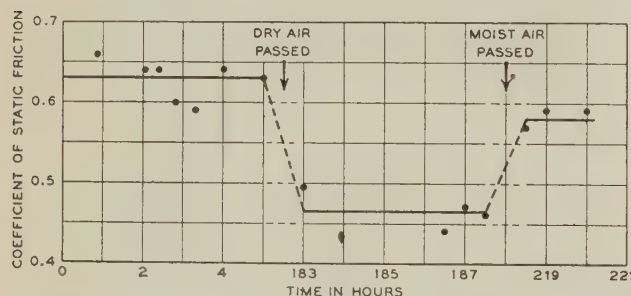


FIG. 2 EFFECT OF VARIATION IN MOISTURE CONTENT OF THE ATMOSPHERE ON STATIC COEFFICIENT OF FRICTION BETWEEN LUBRICATED STEEL SURFACES

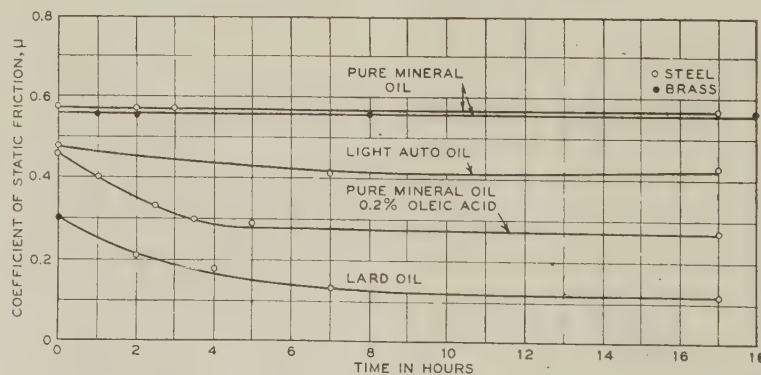


FIG. 3 CHANGE WITH TIME OF COEFFICIENT OF STATIC FRICTION BETWEEN LUBRICATED STEEL AND BRASS SURFACES IN HUMID ATMOSPHERE (Relative humidity, 75 per cent; temperature, 77 F.)

before the surfaces were flooded with oil. Seventy measurements of friction, in groups of ten, were then made over a period of 5 hr, moist air passing continuously. The value thus obtained was 0.63, the dry value for the oil being 0.35. Dry air was then passed through the chamber for 178 hr, when the friction had dropped to 0.49. Further measurements over a period of 4 hr gave a mean of 0.46. Moist air was then passed for 30 hr, after which the friction had risen to 0.57. Two more sets of readings over a period of $1\frac{1}{2}$ hr raised the value to 0.58.

It is clear that the phenomenon is reversible to a considerable extent, though the process of removal of the water is very slow. It is of interest to note that the original high value was never attained on remoistening the air, probably because, over the long periods required to produce drying, some oxidation, resulting in the formation of more polar compounds, took place.

TIME EFFECTS

In many of the runs made on the moist surfaces, a downward trend was noted, and readings were continued until equilibrium was reached. The results of these measurements are plotted in Fig. 3, each point representing the mean of 10 friction readings. Similar results were obtained on a large variety of oils, those plotted in the figure being representative of the behavior observed. The curves are strikingly similar to those for latent period in the studies previously mentioned.

It will be noticed that for mineral oils on brass and steel there is little or no downward trend with time, but that addition of a polar compound not only lowers the initial friction but produces a considerable downward trend. It is also of interest that the friction coefficient, approached as an equilibrium value, is, in general, close to that obtained for the dry oil on dry surfaces, although it was not found to be so closely reproducible.

It is evident that the initial effects are due to a moisture film present at the metal-liquid interface, and it seems a fairly obvious conclusion that the downward trend is, at least in part, due to displacement of this film by polar oil molecules. There is also a possibility, however, that chemical action at the interface, resulting in the formation of soaps, is a complicating factor which may explain the lack of reproducibility of the equilibrium times and values. Although no results are given for glass in Fig. 3, the value was frequently determined after allowing 24 hr for equilibrium to be attained. In only one case was anything but relatively slight lowering found in this time—in one experiment the value for lard oil dropped to 0.24 after 64 hr. This indicates that, given sufficient time, some molecular types may replace the adsorbed moisture film, but it is clear that this film is exceedingly firmly attached. In the case of the white mineral oil, no downward trend of any kind was ever noted.

The results for glass lend support to the idea that formation of

soaps at the metal-oil interface is responsible, at least in part, for the latent-period effect in Fig. 3, since no corrosion reaction would be expected for glass. The relatively long times taken for the steady low value to be reached give further support to the corrosion theory, since the formation of a corrosion film would probably depend upon diffusion of oxygen to the interface. Certainly orientation in a fluid medium would be very rapid and results of preferential wetting studies would also lead to the expectation of relatively rapid effects.

EFFECT OF AGITATING LUBRICATED SURFACES IN A MOIST ATMOSPHERE

Another indication of the mechanism of the moisture effect was provided by agitating the surfaces as described for the graphited oil in Table 3. The results of agitation are given in Table 5.

TABLE 5 EFFECT OF AGITATING MOIST LUBRICATED SURFACES IN MOIST AIR

Lubricant	Surfaces	Initial moist value	Value after agitation μ
White mineral oil.....	Steel	0.58	0.50
White mineral oil.....	Brass	0.56	0.49
White mineral oil.....	Glass	0.93	1.03
Lard oil.....	Steel	0.30	0.11
Lard oil.....	Glass	0.85	0.80
Oleic acid.....	Steel	0.20	0.12
Oleic acid.....	Brass	0.34	0.13
Oleic acid.....	Glass	0.81	0.75

Here again, the effects are completely consistent with preferential wetting measurements, if it is assumed that disturbance of the interface layers assists the attainment of adsorption equilibrium. It will be noticed that a sharp drop in friction takes place after agitation in the case of the metals, the coefficient dropping to a value close to the dry value for strongly adsorbed film types, and to an intermediate value for the mineral oils. For the white mineral oil, the drop was temporary, the value rising again to the initial high value on standing. Further information is provided in this table to justify the assumption of a persistent adsorbed water film on glass, the friction being but slightly affected by agitation in any case tested.

In Table 6 are given some rather interesting and unexpected

TABLE 6 EFFECT OF AGITATING DRY LUBRICATED SURFACES IN MOIST AIR

Lubricant	Surfaces	Dry value μ	Value after agitation
White mineral oil.....	Brass	0.30	0.42
Light auto oil.....	Steel	0.32	0.41
Spindle oil + 2 per cent oleic acid.....	Steel	0.18	0.29
Oleic acid.....	Steel	0.12	0.17
Castor oil.....	Steel	0.15	0.24

data obtained when the surfaces were agitated in an atmosphere of 50 per cent relative humidity after an equilibrium value had been obtained for the dry oil on the dry surfaces. It will be noticed that for every combination tested, a significant rise in friction was observed. This rise was not persistent, the value of μ dropping to its original equilibrium value after a few hours. The rise can be explained on the assumption that the agitation produces a temporary change in the adsorption equilibrium, moisture from the air being enabled to occupy temporarily some positions in the interfacial film. The subsequent drop is probably generally due to formation of metal soaps at the interface.

RELATION BETWEEN FRICTION AND MOLECULAR WEIGHT OF LUBRICANT

Having determined the nature and magnitude of the moisture effects to be observed, a series of further runs was made to attempt to reproduce the linear relation found by other workers

(6, 9) to hold between the coefficient of static friction and the molecular weight for a given homologous series. Samples of fatty acids were carefully purified by redistillation in vacuum until their boiling points checked those given in International Critical Tables to within ± 0.1 deg C. In order to approximate the conditions used in the studies mentioned, watch-glass sliders weighted with lead were used in this work. The sliders and glass plate were cleaned by immersion in warm chromic acid, thorough scrubbing with cotton under tap water, washing with redistilled alcohol, and drying in a current of dry air. The experiments were carried out in a chamber which was kept more carefully dried than in the experiments already described. The slider and plate were dried for $1\frac{1}{2}$ hr in the chamber out of contact with each other. The lubricant was added in such a manner that no moisture could be absorbed during addition, and was spread over the plate with a clean glass rod inserted in the exit hole of the chamber. The results are given in Table 7.

TABLE 7 COEFFICIENT OF STATIC FRICTION BETWEEN GLASS SURFACES LUBRICATED BY FATTY ACIDS

Fatty acid	No. of atoms	μ	Average deviation from mean, per cent	μ Hardy and Doubleday
Propionic.....	3	0.57	11.1	0.63
		0.56	12.7	
Valeric.....	5	0.48	9.9	0.53
		0.60	13.8	
Pelargonic.....	9	0.44	6.5	0.28

These results, in spite of the extreme precautions taken to insure cleanliness and dryness of the atmosphere, show a wider spread than many obtained under much less carefully controlled conditions. It is believed that this is in part due to the inherent instability of a spherical slider; it was impossible to apply the load without causing rocking. It is of interest that no trend downward was observed in these experiments, which lends more support to the idea that latent period is due to moisture and corrosion effect.

Although the results are not as complete as could be desired, it was not felt worth while to make further measurements on any other members of the series. It is quite clear that there is no simple linear relationship between coefficient of friction and molecular weight under the conditions of test, nor is the reproducibility sufficient to draw more than a qualitative conclusion. All that can be stated is that there is a tendency for the friction coefficient to decrease with increase in the molecular weight of the fatty acids and paraffins. This is in agreement with the dynamic-friction results of Beare and Bowden (12) who found no simple linear relationship between the coefficient of friction and molecular weight.

DISCUSSION

The phenomenon of solid friction has been explained as being due to the interlocking of surface asperities (13). Another view is that friction is due to the attractive forces between the metal atoms across the interface (14). Although it is probable that the interlocking of surface asperities plays some part in frictional processes, particularly when dynamic conditions prevail, it is obvious that the moisture effects observed in this study cannot be explained by such a mechanism. On the molecular theory of friction, the moisture effects can be explained by assuming that the lubricant molecules are more weakly bonded and less strongly oriented at an adsorbed water surface. Under these conditions the metal atoms separated by the boundary film approach one another more closely. The increase in friction for lubricated moist surfaces would then be due to the additional force needed to separate a larger number of metal bonds.

Although corrosion of the metal, resulting in the formation of

an oriented soap film at the metal-oil interface, is a likely cause of the downward trend with time, observed for polar-type lubricants on moist surfaces, it is not believed possible that the low results obtained under dry conditions for nonpolar oils on metals, and for all the oils on glass, can be explained in this way.

Many of the effects recorded in this paper may be modified for surfaces undergoing continuous relative motion, as indeed the results of agitation would lead one to expect. There is little doubt, however, that under certain conditions, such as those prevailing when a mechanism is started on a damp day, adsorption of water on the metal surface can exercise an important effect. Similar studies of the effect of moisture on dynamic friction and wear between lubricated metal surfaces should be of great fundamental interest.

BIBLIOGRAPHY

- 1 "On the Lubricating and Other Properties of Thin Oily Films," by Lord Rayleigh, *Philosophical Magazine*, series 6, vol. 35, 1918, pp. 157-163.
- 2 "Boundary Lubrication—The Latent Period and Mixtures of Two Lubricants," by W. B. Hardy and I. Doubleday, *Proceedings of the Royal Society of London*, series A, vol. 104, 1923, pp. 25-38.
- 3 "Lubricating Properties of Greases From Petroleum Oils," by F. H. Rhodes and H. D. Allen, *Industrial and Engineering Chemistry*, vol. 25, 1933, pp. 1275-1280.
- 4 "Studies in Boundary Lubrication," by W. E. Campbell, *Trans. A.S.M.E.*, vol. 61, 1939, pp. 633-641.
- 5 "The Analysis of Commercial Lubricating Oils by Physical Methods," by W. B. Hardy and M. Nottage, *Lubrication Research*, Technical Paper No. 1, Department of Science and Industrial Research, H. M. Stationery Office, Great Britain, 1929.
- 6 "Boundary Lubrication—The Paraffin Series," by W. B. Hardy and I. Doubleday, *Proceedings of the Royal Society of London*, series A, vol. 100, 1921-1922, pp. 550-574.
- 7 "Spreading of Liquids on Solid Surfaces," by R. Bulkley and G. H. S. Snyder, *Journal of the American Chemical Society*, vol. 55, 1933, pp. 194-208.
- 8 "Quantitative Correlation of Interfacial Free Surface Energies," by F. E. Bartell and L. S. Bartell, *Journal of the American Chemical Society*, vol. 56, 1934, pp. 2205-2210.
- 9 "Boundary Lubrication—The Temperature Coefficient," by W. B. Hardy and I. Doubleday, *Proceedings of the Royal Society of London*, series A, vol. 101, 1922, pp. 487-492; also see ref. 6.
- 10 "Boundary Lubrication—Further Consideration of the Influence of the Composition of the Solid Face," by I. Doubleday, *Proceedings of the Royal Society of London*, vol. 106, 1924, p. 341.
- 11 "A Study of the Boundary Lubricating Value of Mineral Oils of Different Origin," by M. E. Nottage, *Lubrication Research*, Department of Science and Industrial Research, Technical Paper No. 2, H.M. Stationery Office, Great Britain, 1934.
- 12 "Physical Properties of Surfaces. I. Kinetic Friction," by W. G. Beare and F. P. Bowden, *Philosophical Transactions of the Royal Society of London*, series A, vol. 234, 1935, pp. 329-354.
- 13 "Histoire de l'Académie des Sciences avec les Mémoires de Mathématique et de Physique, by Amontons, 1699, p. 206; also "Mémoires de Mathématique et de Physique de l'Académie Royale des Sciences," by C. A. Coulomb, 1785, p. 161.
- 14 "Problems of the Boundary State," by W. B. Hardy, *Philosophical Transactions of the Royal Society of London*, series A, vol. 230, 1932, pp. 1-37.
- 15 "A Molecular Theory of Friction," by G. A. Tomlinson, *Philosophical Magazine*, series 7, vol. 7, 1929, pp. 905-939.
- 16 "The Nature of Friction," by P. E. Shaw, *Philosophical Magazine*, series 7, vol. 9, 1930, pp. 628-629.

Discussion

J. J. BIKERMAN.⁵ The authors observed an increase in frictional force on adding small amounts of water to two lubricated hydrophilic solid surfaces in contact. They state, "this high friction cannot be due to ordinary capillary forces" and, "the moisture effects can be explained by assuming that the lubricant

molecules are more weakly bonded and less strongly oriented at an adsorbed water surface." The writer is unable to agree with either of these statements.

A convincing interpretation of the authors' experimental results is rendered particularly difficult because they did not check whether or not the law of friction was valid in their work. They do not mention having varied the load on their slider or the apparent area of contact between slider and support. If the law of friction were valid, the cause would be sought in the roughening produced by alternate moistening and drying. If the law of friction were not valid, then "ordinary capillary forces" presumably were responsible for the enhanced friction.

B. O. W. L. Ljunggren⁶ describes a clear case of capillary attraction affecting sliding. He pulled a diamond splinter loaded with about 49 dynes along a polished iron surface. Microscopic scratches were produced in this manner. When the surface was lubricated with neatsfoot oil, the scratches became wider because the oil surface between diamond and steel tended to contract. A separate test showed that the capillary attraction was as great as 68 dynes and that the width of the scratches corresponded to the load increased by this attraction.

There exists another capillary phenomenon which usually affects frictional force more than does capillary attraction. It is due to hysteresis of wetting. It was studied, for instance, by G. D. West,⁷ V. S. Veselovskii and V. N. Pertsov,⁸ J. J. Bikerman,⁹ and others. The force required to push a drop along a surface poorly wetted by the liquid is greater the greater the drop, and the greater the hysteresis of contact angle. Quantitative statements cannot be made as long as so little is known about the capillary properties of the system studied by the authors.

The interpretation given by the authors is a good example of the vagueness of the molecular theory of friction. The authors assume that in their experiments a water film is present on the solid surface and an oil film is present on top of the water film. They state: "Under these conditions the metal atoms separated by the boundary film approach one another more closely." If water had reduced the frictional force, the explanation would have been that a double film (i.e., water + oil) obviously is thicker than the oil film alone. Under these conditions, the metal atoms separated by the boundary film approach one another less closely. A theory which, with equal facility, can be made to agree with contrary experiments cannot be considered useful.

The writer was pleased to see that the authors' experiments contradict Hardy's belief in a strict correlation between the molecular weight and the efficiency of a boundary lubricant. This correlation was Hardy's main argument for the molecular theory of friction. A criticism of this theory can be found in a paper by the writer.¹⁰

M. E. MERCHANT.¹¹ Clearly the authors have tracked down one of the basic causes of disagreement and lack of reproducibility.

⁶ "Method of Sclero-Grating Employed for the Study of Grain Boundaries and of Nitrided Cases; Grain Structures Revealed by Cutting," by B. O. W. L. Ljunggren, *Journal of the Iron and Steel Institute*, vol. 141, 1940, pp. 341P-404P.

⁷ "On the Resistance to the Motion of a Thread of Mercury in a Glass Tube," by G. D. West, *Proceedings of the Royal Society of London, England*, vol. 86, 1911-1912, pp. 20-25.

⁸ "Adhesion of Drops to Solid Surfaces," by V. S. Veselovskii and V. N. Pertsov, *Journal of Physical Chemistry (USSR)*, vol. 8, 1936, pp. 245-259.

⁹ "On the Formation and Structure of Multilayers," by J. J. Bikerman, *Proceedings of the Royal Society of London, England*, vol. 170, 1939, pp. 130-144.

¹⁰ "Surface Roughness and Sliding Friction," by J. J. Bikerman, *Review of Modern Physics*, vol. 16, 1944, pp. 53-68.

¹¹ Research Physicist, The Cincinnati Milling Machine Company, Cincinnati, Ohio. Mem. ASME.

⁵ Woodside, Long Island, N. Y.

bility in many of the boundary-lubrication studies made by others in the past. They have also made a real contribution to our knowledge of the behavior of adsorbed films and their role in boundary friction. It is such studies as these which are needed to solve the many knotty riddles which still prevent us from arriving at a harmonious picture of the mechanisms of boundary lubrication. We hope that the authors will continue to push back these frontiers.

The authors' results seem clearly to establish the fact that the high friction values which they find in certain cases are due to the presence of an adsorbed film of moisture on the solid surfaces; there appears to be no other logical explanation. The findings for the case where glass friction surfaces were used are particularly significant, since in this case the possibility of the effects being due to chemical reaction with the surfaces, to form oxides or other solid films of reduced shear strength, is practically excluded. Further, the facts that (a) the adsorbed water molecules can "block off" the adsorption of the long oily molecules on a surface and thus keep friction at a high value, and (b) they can do so reversibly, are also important. These findings, too, seem to indicate quite clearly that the mechanism of friction reduction by the oily molecules, under the conditions of the present tests, is that of physical adsorption of the liquid lubricant rather than that of chemical reaction of the surface and lubricant to form protective films of solid material. On the other hand, friction studies made in recent years by certain other investigators^{12,13} have indicated that, under the conditions of their tests, effective boundary lubrication is obtained only when the two friction surfaces are kept apart by a film of low-shear-strength solid material; adsorbed films of chemically unchanged liquids were found to be ineffective. Thus the present study again turns the spotlight on this basic riddle of boundary friction, still unsolved, namely, that of the role of adsorbed liquid films versus protective solid films as friction-reducing agents, and the mechanism and conditions under which each of these can function effectively. When can adsorbed liquid films reduce friction, and how, and what conditions are necessary to make them ineffective so that solid films must be used? Is it, perhaps, that the presence of moisture is the condition necessary to make adsorption ineffective? It is hoped that the authors may be able to throw some light on this basic problem in the future studies.

W. A. ZISMAN.¹⁴ The authors have exercised unusual care in selecting and controlling the conditions for their measurements of the coefficient of static friction, and the result is a valuable addition to the scientific literature on the subject of boundary lubrication and adsorbed films. The importance of the effect of adsorbed water is demonstrated beyond doubt, and the large effects reported should interest and warn future investigators of boundary-lubrication phenomena.

The effect of water on hydrocarbon films (Tables 2 and 3 of the paper) appears to be caused by a displacement of the adsorbed hydrocarbon molecules by the more polar molecules of water. This is related to the well-known fact that a layer of a pure hydrocarbon will not long protect iron from the rusting effects of humid air. The values of μ of 0.75 (in presence of dry air) and 1.05 (in moist air) for unlubricated steel on steel dropped to 0.35 and 0.60, respectively, resulting in decreases in μ of 0.40 and 0.45. This nearly constant decrease in μ may be a measure

of the ability of a weakly adsorbed film of hydrocarbon molecules to lessen metal-to-metal contact under a shearing stress. That the coefficient of friction of a slider in a pool of hydrocarbon oil is lower in the presence of dry air than moist air can be explained by the greater ease of displacing a hydrocarbon layer from a water-covered surface than from a dry metallic surface. After all, the thin layer of water will offer much less resistance to a shearing stress than a layer of metal or of a metallic oxide. The hydrocarbon film will rest on a firmer foundation when resting on a dry metal, and hence it will not rub off so readily and so will be more effective in decreasing the amount and duration of metal-to-metal contacts.

In the absence of oil, the observed coefficient of friction for dry steel on steel is probably that for iron covered with a monolayer or more of an oxide of iron. When water is adsorbed, the wet film of iron oxide may become transformed to a hydrated iron oxide which may shear more readily under stress. That would lead to the observed higher coefficient of static friction resulting from the adsorption of water. If that adsorbed water is partly or entirely removed by a current of dry air, the film left behind may revert to the mechanically stronger form. Such a mechanism may be the basis of the partially reversible effects discussed by the authors in connection with their Fig. 2. An adsorbed film may be thin, but like matter in bulk it may undergo chemical or physical transformations which greatly alter its boundary properties. It would appear valuable to obtain electron-diffraction data before a satisfactory theory of the phenomena reported by the authors can be established.

In connection with Fig. 3 of the paper, the slow decreases in μ observed with mineral oil plus 0.2 per cent oleic acid and with lard oil have been attributed by the authors to the formation of a corrosion film. It is more likely that these observations are caused by the adsorption of a monolayer of a very adsorbable impurity present in the oil (or additive) in a concentration of less than 1 part in 10,000. Due to the viscosity of the oil through which the polar molecules must diffuse to reach the steel surface where rubbing occurs, still more time is required for formation of a close-packed monolayer. This would explain why agitating the surfaces (see Table 5) causes more rapid attainment of the equilibrium condition of the film.

The several possibilities in the interpretation of the authors' data are indicative of the difficulties of obtaining rigorous conclusions about boundary lubrication in the present state of knowledge of the condition of films adsorbed at the oil-metal and water-metal interfaces. The authors' painstaking studies of static friction are timely and suggestive. If further work is done it is hoped that equally careful measurements of μ will be made on pure hydrocarbons containing known and controlled solutions of pure polar compounds, that other polar materials than acids be tried, and that electron-diffraction observations be taken of the steel surfaces at different stages of the experiments.

AUTHORS' CLOSURE

By ordinary capillary forces, the authors meant those at the surface of a mobile liquid. In the experiments described the films cannot be more than a few molecular layers thick; they may be monomolecular. The forces involved in separating these layers from the solid probably contribute largely to the friction.

Mr. Bickerman criticizes the molecular theory of friction by interpreting it in his own way for a hypothetical case, and taking this interpretation as proof of the theory's lack of usefulness. The authors do not find this criticism very convincing.

The friction between unlubricated steel surfaces was the same whether the surrounding atmosphere was dry or moist. In fact, value of μ of 0.78 was obtained when the surfaces were flooded with water. The values of μ for steel—steel above 0.78 were ob-

¹² "Lubrication of Metal Surfaces by Fatty Acids," by F. P. Bowden, J. N. Gregory, and D. Tabor, *Nature*, vol. 156, 1945, pp. 97-101.

¹³ "Characteristics of Typical Polar and Non-Polar Lubricant Additives Under Stick-Slip Conditions," by M. E. Merchant, *Lubrication Engineering*, vol. 2, June, 1946, pp. 3-8.

¹⁴ Chief, Lubrication Section, Naval Research Laboratory, Washington, D. C. Mem. ASME.

tained when the surfaces were brought together in a moist atmosphere which was subsequently dried. These high values were transient, and depended upon the adhesive effect of a water layer trapped at the steel, i.e., steel interface. Dr. Zisman's explanation for the reversible effects, illustrated in Fig. 2 of the paper is therefore not in accord with experiment.

The formation of soaps at the interface is favored as an explanation of the trends in Fig. 3 over Dr. Zisman's explanation, because similar trends were not observed when the same oils were measured in a dry atmosphere. Furthermore, in some experiments with brass, a green color was observed to develop, with time, over the metal surface.

Measurements of Combined Frictional and Thermal Behavior in Journal-Bearing Lubrication

By S. A. McKEE,¹ H. S. WHITE,² AND J. F. SWINDELLS²

Data were obtained in tests with a four-bearing friction machine which show that an increase in the load on a journal bearing produces a proportional increase in frictional torque when other conditions of test are held constant. Under these same conditions an increase in load also produces a proportional increase in the fluidity of the oil in the bearing. These two effects are the result of the combined hydrodynamic and thermodynamic actions involved in journal-bearing operation with forced-feed lubrication. The increase in torque is influenced by the viscosity of the oil, the oil-inlet temperature, the oil-feed pressure, the shaft diameter, the clearance-diameter ratio, and the length-diameter ratio. The increase in fluidity is influenced by the same factors and also by the speed of the journal. Empirical equations are derived for the fluidity-pressure relationship for certain conditions. Also, a graphical method is given for the use of this relationship in estimating safe bearing loads.

NOMENCLATURE

The following nomenclature is used in the paper:

- D = journal diameter, in.
- L = bearing length, in.
- C = running clearance (difference between bearing diameter and journal diameter), in.
- W = total load acting on bearing, lb
- $P = W/LD$ = pressure on projected area of bearing, psi
- N = speed of journal, rpm
- F = tangential frictional force, lb
- $f = F/W$ = coefficient of friction
- T = frictional torque, lb-in.
- Z = absolute viscosity of lubricant at atmospheric pressure and bearing temperature, centipoises
- $1/Z$ = fluidity of lubricant at atmospheric pressure and bearing temperature, reciprocal centipoises
- $1/Z_0$ = fluidity of lubricant at atmospheric pressure and oil-inlet temperature, reciprocal centipoises
- $1/Z'$ = intercept [at $PND (L/D) (D/C) = 0$] of line representing relation between $1/Z$ and $PND (L/D) (D/C)$.
- a = constant in Equations [1] and [2]
- b = constant in Equation [1]
- m = constant in Equation [2]
- n = exponent in Equation [2]
- H_t = rate of heat supplied to one bearing from both entering oil and bearing friction, in-lb/min.

ΔT_b = temperature rise above ambient of loaded side of bearing shell (average of four bearings), deg F

ΔT_o = temperature rise above ambient of oil leaving ends of bearings (average of four bearings), deg F

Q = rate of oil flow per bearing, cu in. per min

ZN/P = generalized operating variable

INTRODUCTION

The operation of a journal bearing under a steady state of temperature distribution with forced-feed lubrication is dependent upon a number of factors pertaining to the conditions of operation and to the geometry of the bearing. The relation of these factors to the performance of the bearing is very complex, involving both the rate at which the heat is generated in the bearing and the rate at which it is carried away. This paper deals with the general characteristics of data observed in individual test runs with journal bearings and involves an analysis of the resultant effects of the combined hydrodynamic and thermodynamic actions upon the performance of the bearings.

These tests were part of a research program carried out with the co-operation and financial assistance of the National Advisory Committee for Aeronautics. They were made with a four-bearing friction machine and covered a wide range of conditions typical of heavy-duty service.

APPARATUS

Four-Bearing Friction Machine. The apparatus used in this investigation has been described in detail in a previous publication.³ The four-bearing friction machine and auxiliary apparatus are shown in Fig. 1. Fig. 2 shows the major elements of the friction machine disassembled. The machine consists essentially of four test bearings on a common shaft. The bearings are mounted in self-aligning ball-bearing swivels which are prevented from rotating by flat springs. The two outer bearings are fastened to the housing while the two inner ones are located in plates sliding in guides. Load is applied by hydraulic jacks under the two inner bearings. The complete housing floats on the horizontal shaft and acts as a cradle dynamometer.

Oil is fed to the bearings through the drilled shaft which has two oil holes at the longitudinal center of each bearing. Oil-feed pressure is maintained by a motor-driven pump with a relief valve for pressure control.

Bearing temperatures are measured by thermocouples embedded in the metal at the loaded sides of the bearings.

Shafts and Bearings. The shafts used in these tests were made of crankshaft steel. Two nominal sizes were used, 2 in. diam and 1 1/4 in. diam.

The test bearings were solid steel sleeves lined with a thin layer of copper-lead. Their nominal dimensions were as follows:

³ "Performance Characteristics of Journal Bearings With Forced-Feed Lubrication," by S. A. McKee, H. S. White, A. D. Bell, and J. F. Swindells, NACA Wartime Report ARR No. 4H15, Aug., 1944.

¹ Chief, Lubrication Section, National Bureau of Standards, Washington, D. C.

² Associate Physicist, National Bureau of Standards.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 47-A-61.

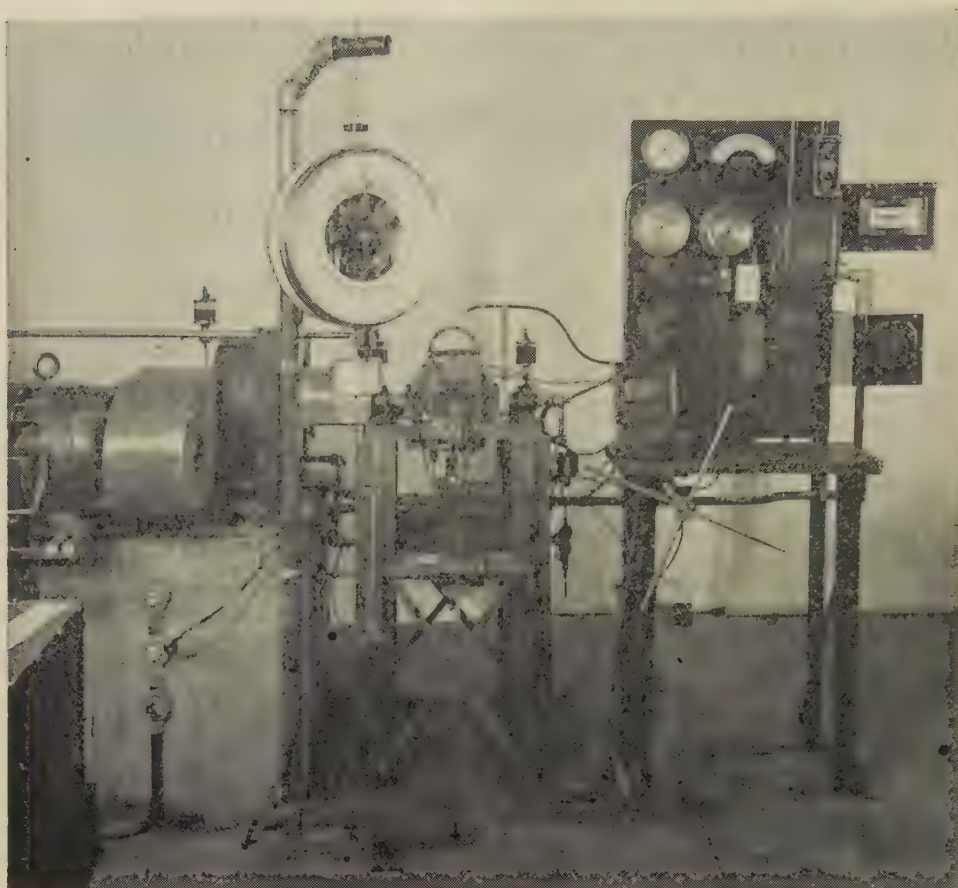


FIG. 1 FRICTION-MACHINE INSTALLATION

3 sets of 2-in-diam \times $1\frac{1}{4}$ -in. bearings having D/C ratios (see nomenclature) of 1142 (set 1a), 556 (set 1b) and 343 (set 1c), respectively.

3 sets of $1\frac{1}{4}$ -in-diam \times $25/32$ -in. bearings having D/C ratios of 1042 (set 2a), 500 (set 2b), and 338 (set 2c), respectively.

2 sets of $1\frac{1}{4}$ -in-diam \times $1\frac{1}{4}$ -in. bearings having D/C ratios of 500 (set 3a) and 329 (set 3b), respectively.

Lubricants. The lubricants used were a Pennsylvania oil (NBS laboratory reference number J3-120), a Navy Contract 1080 oil, and an SAE 20 oil. The viscosity data for these oils are given in Table 1.

TABLE 1 TEST OILS, VISCOSITY DATA

Lubricant	Viscosity			
	Saybolt sec		Centistokes	
	100 F	210 F	100 F	210 F
NBS Reference J3-120.	1766	124.4	382.3	26.04
Navy Contract 1080...	781	77.5	169.1	14.95
SAE 20 used with 2-in. bearings.....	371	58.6	80.2	9.82
SAE 20 used with $1\frac{1}{4}$ -in. bearings.....	382	58.9	82.58	9.90

Test Runs. In these tests the bearings were operated under conditions of stable lubrication at the higher values of ZN/P (see nomenclature). Test runs were made at constant speeds and at a number of constant loads which were successively increased during each test run. The data were obtained with the apparatus thoroughly "warmed up" and with the bearings in a steady state of temperature distribution.

In general, the tests with the eight sets of bearings covered operation with all three grades of oil; at three oil-inlet temperatures, 150 F, 200 F, and 250 F; a nominal oil-feed pressure of 35

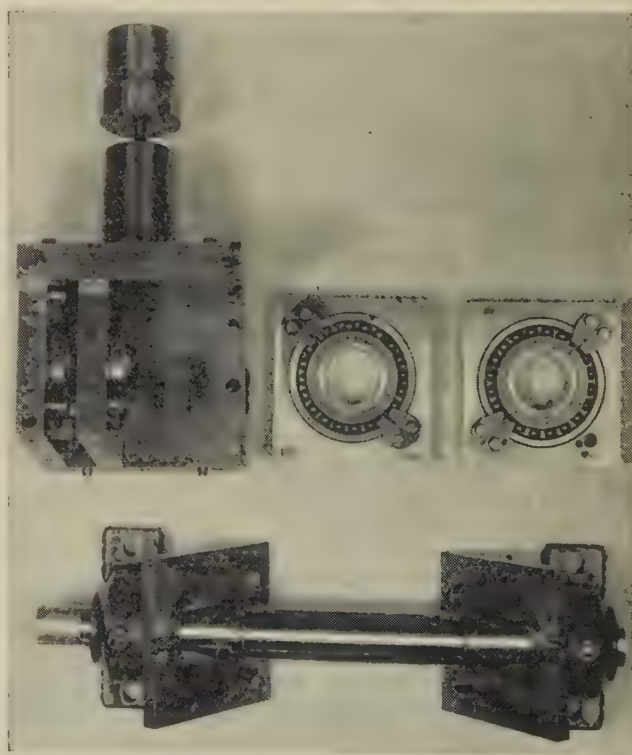


FIG. 2 FRICTION-MACHINE PARTS

psi; and two nominal speeds of 2000 and 3000 rpm. Tests were also run at other oil-feed pressures using the 1080 oil at 200 F oil-inlet temperature and a journal speed of 2000 rpm. With set 2a, however, all the tests with the 1080 oil were omitted. The range of loads covered in a given test was dependent upon the particular conditions present. The minimum load used was 273 psi on the projected area (with sets 3a and 3b), while the maximum was 4327 psi on the projected area (with set 2c). The bearing temperatures ranged from 163 F to 343 F. Ambient temperatures averaged slightly over 80 F.

TYPICAL TEST DATA

Data obtained in three typical test runs while operating at a given constant speed and various loads are shown graphically in Figs. 3, 4, and 5. In these figures, observations of the frictional torque, average bearing temperature, and rate of oil flow are plotted against the load expressed in pressure on the projected area. Data are also shown for the coefficient of friction, absolute viscosity of the lubricant at bearing temperature, and fluidity of the lubricant (reciprocal of viscosity) for each load. The values for the viscosity and fluidity of the lubricant are based on the observed bearing temperature.

From these figures, it will be noted that the individual variables show about the same general trends with change in load for each of the three tests covering two diameters, D/C ratios, L/D ratios, and speeds, and three viscosity grades and oil-inlet temperatures.

FRICTIONAL TORQUE DATA

One of the general characteristics of interest, shown in Figs. 3, 4, and 5, is the relation between frictional torque and load.

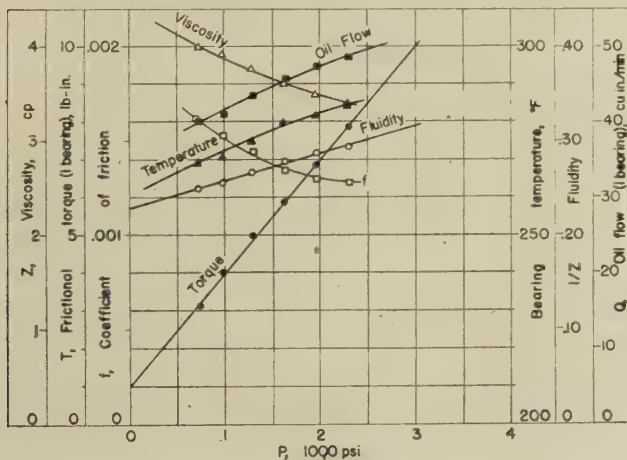


FIG. 3 TYPICAL TEST DATA FOR BEARING SET 1b
(2-in. \times 1 1/4-in. copper-lead bearings, $L/D = 0.620$, $D/C = 556$, 3000 rpm, SAE 20 oil at 250 F oil-inlet temperature, and 32 psi average oil pressure.)

When a given bearing is operating at a constant speed, using a given oil at constant oil-inlet temperature and constant oil-feed pressure, an increase in load produces an approximately proportional increase in frictional torque. This straight-line relationship holds reasonably well in all the tests with the eight sets of bearings involving operation at two speeds, three oils, and three oil-inlet temperatures, a total of about 150 test runs covering a fairly wide range of conditions.

Torque data covering a range of operating conditions with bearing sets 1b and 3a are given in Figs. 6 and 7, respectively. Consideration of the intercepts (at $P = 0$) of the torque versus pressure curves in these figures indicates that a decrease in viscosity either by the use of a different oil or by an in-

crease in oil-inlet temperature decreases the torque at no load. Since these intercepts represent approximately conditions of concentric running, this trend might be expected from the Petroff equation.⁴ It should be noted, however, that these data also involve the temperature rise in the bearings. This is illustrated with the effect of change in speed when using the same oil at the same oil-inlet temperature. In general, the increase in no-load torque is smaller proportionally than the increase in speed, because of the higher temperature and consequent lower viscosity at the higher speed.

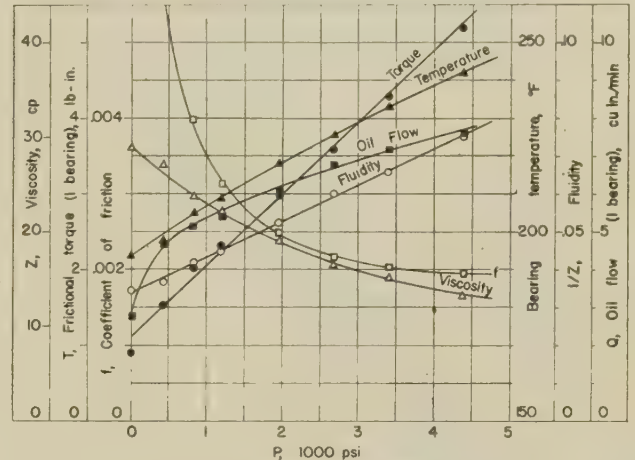


FIG. 4 TYPICAL TEST DATA FOR BEARING SET 2c
(1 1/4-in. \times 25/32-in. copper-lead bearings, $L/D = 0.633$, $D/C = 338$, 3050 rpm, J3-120 oil at 150 F oil-inlet temperature, and 37 psi average oil pressure.)

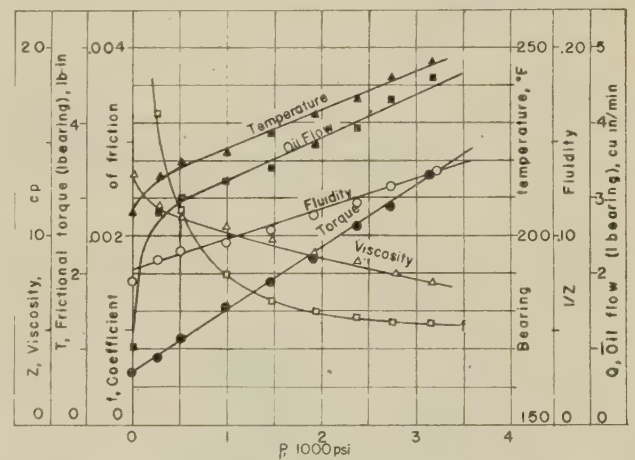


FIG. 5 TYPICAL TEST DATA FOR BEARING SET 3a
(1 1/4-in. \times 1 1/4-in. copper-lead bearings, $L/D = 1.002$, $D/C = 500$, 2080 rpm, 1080 oil at 200 F oil-inlet temperature, and 34 psi average oil pressure.)

When the slopes of these curves (increase in torque per unit increase in load) are considered, an analysis of the data for all the bearings tested indicates a general trend for a slight decrease in slope with decrease in viscosity or with increase in oil-inlet temperature. The effect of speed over the range covered (2000-3000 rpm) is practically negligible.

The effects of clearance-diameter ratio, length-diameter ratio, and shaft diameter are shown by the curves in Fig. 8. In this figure, torque versus pressure data are given for sets 1a,

⁴ "Friction in Machines and the Effect of the Lubricant," by N. Petroff, (a) In Russian: Engineering Journal, St. Petersburg, 1883; no. 1, pp. 71-140; no. 2, pp. 228-279; no. 3, pp. 377-436; no. 4, pp. 535-564. (b) German translation, by L. Wurzel, Hamburg; L. Voss, 1887, 187 pp.

1c, 2a, 2c, and 3b when operating with the J3-120 oil at 200 F oil-inlet temperature, and at 2000 rpm or at 2080 rpm. These data are also shown in Fig. 9, where F , the bearing friction ($2T/D$), is plotted against W , the total load on the bearing (PLD). In this figure, the data at $W = 0$ give an indication of the effects of D/C , L/D , and D on the no-load friction for the given operating conditions. The order in which the values occur indicates the general trend of the order of the no-load friction data for these bearings with other oils and at other oil-inlet temperatures.

When considering operation under load, a comparison of

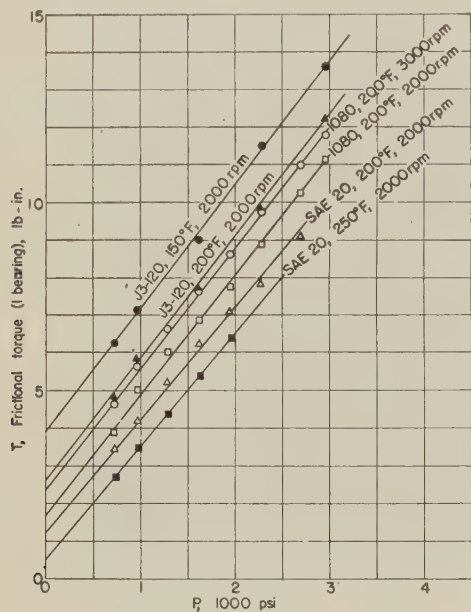


FIG. 6 TORQUE DATA FOR A RANGE OF OPERATING CONDITIONS WITH BEARING SET 1b

(2-in. \times 1 1/4-in. copper-lead bearing, $L/D = 0.620$, $D/C = 556$, at 35 psi average oil pressure.)

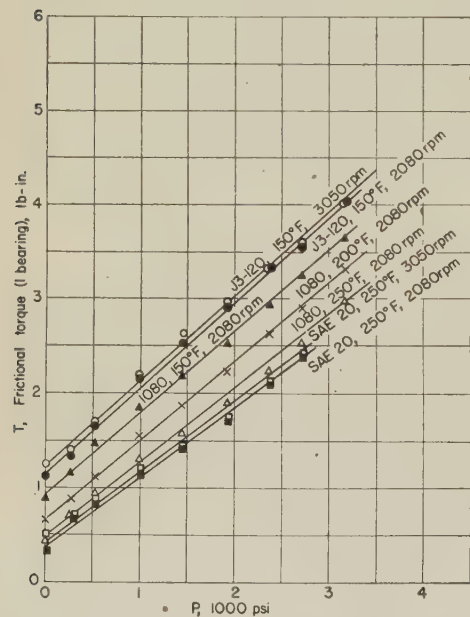


FIG. 7 TORQUE DATA FOR A RANGE OF OPERATING CONDITIONS WITH BEARING SET 3a

(1 1/4-in. \times 1 1/4-in. copper-lead bearings, $L/D = 1.002$, $D/C = 500$, at 34 psi average oil pressure.)

curves 1c with 1a, and 2c with 2a in Fig. 9, indicates that, under the conditions present in these tests, the increase in friction per unit increase in load was greater for the bearings having the larger clearance. In this connection it is of interest that at the higher loads the bearings having the smaller clearance had the lower friction. With a journal bearing operating with forced-feed lubrication at a given load and speed in the region of stable lubrication, the effect of a reduction in clearance is twofold.

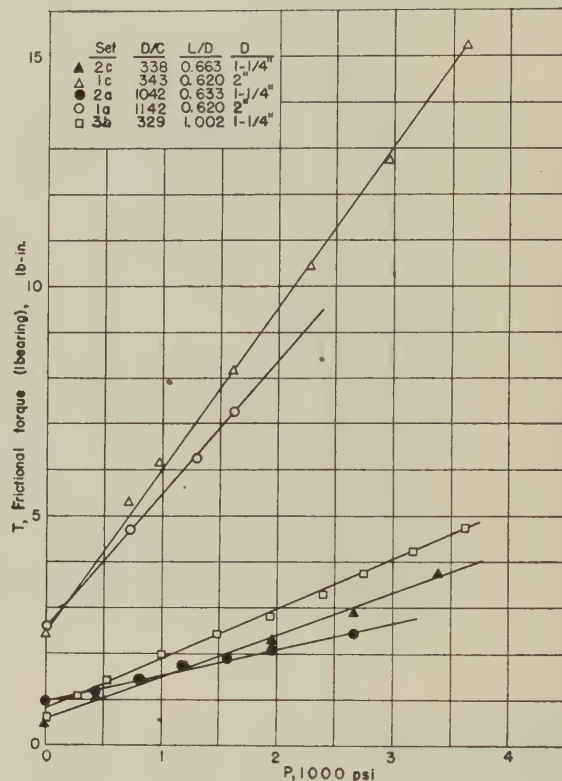


FIG. 8 TORQUE DATA, SHOWING EFFECT OF SHAFT DIAMETER, DIAMETER-CLEARANCE RATIO, AND LENGTH-DIAMETER RATIO

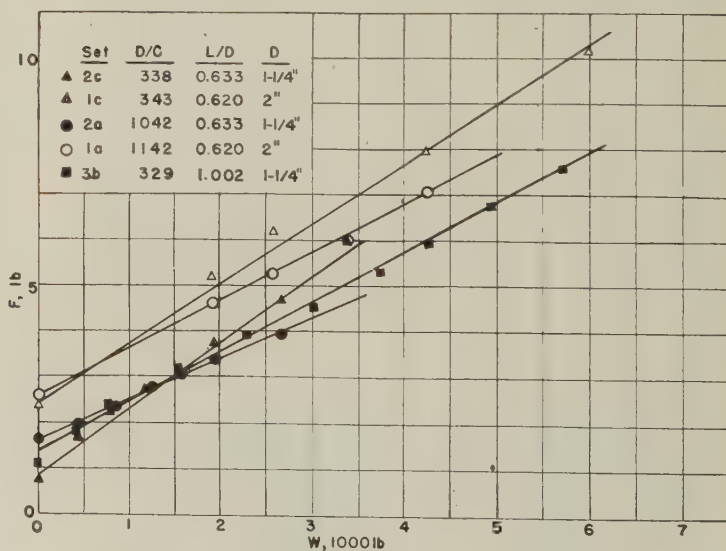


FIG. 9 FRICTIONAL FORCE DATA, SHOWING EFFECT OF SHAFT DIAMETER, DIAMETER-CLEARANCE RATIO, AND LENGTH-DIAMETER RATIO AT 2000 OR 2080 RPM, WITH J3-120 OIL AT 200 F OIL-INLET TEMPERATURE, AND 36 PSI AVERAGE OIL PRESSURE

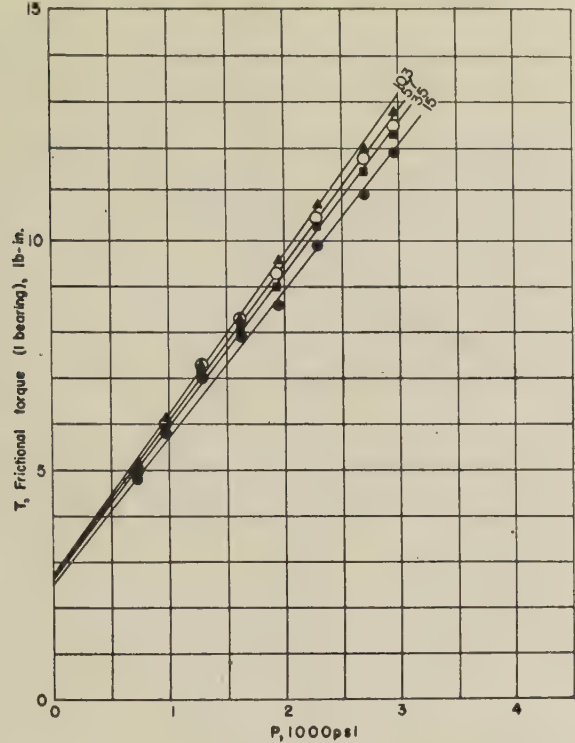


FIG. 10 TORQUE DATA, SHOWING EFFECT OF OIL-FEED PRESSURE WITH BEARING SET 1b
(2-in. \times 1 $\frac{1}{4}$ -in. copper-lead bearings, $L/D = 0.620$, $D/C = 556$, 2000 rpm, 1080 oil at 200 F oil-inlet temperature. Figures at ends of curves indicate oil-feed pressure, psi.)

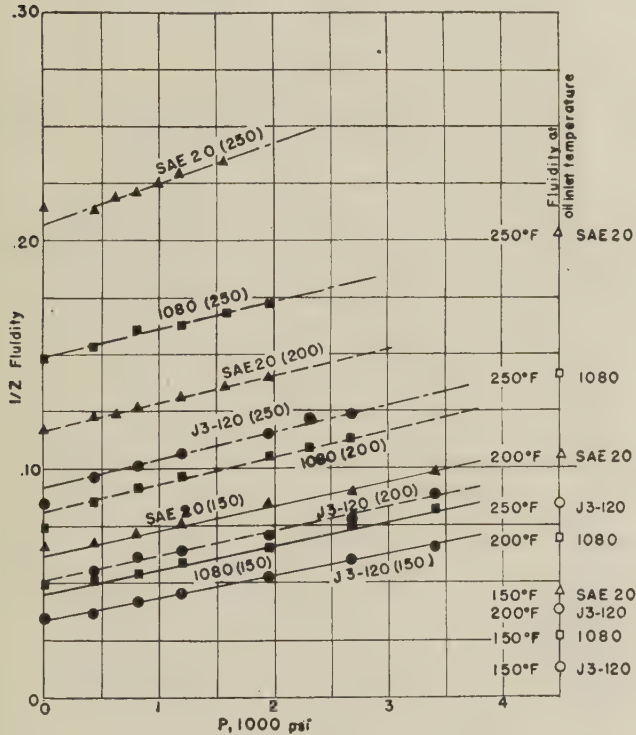


FIG. 12 DATA SHOWING EFFECTS OF VISCOSITY GRADE AND OIL-INLET TEMPERATURE ON FLUIDITY-PRESSURE CHARACTERISTICS WITH BEARING SET 2c
(1 $\frac{1}{4}$ -in. \times 25/32-in. copper-lead bearings, $L/D = 0.633$, $D/C = 338$, 3050 rpm, 36 psi average pressure. Figures in parentheses indicate oil-inlet temperature, deg F.)

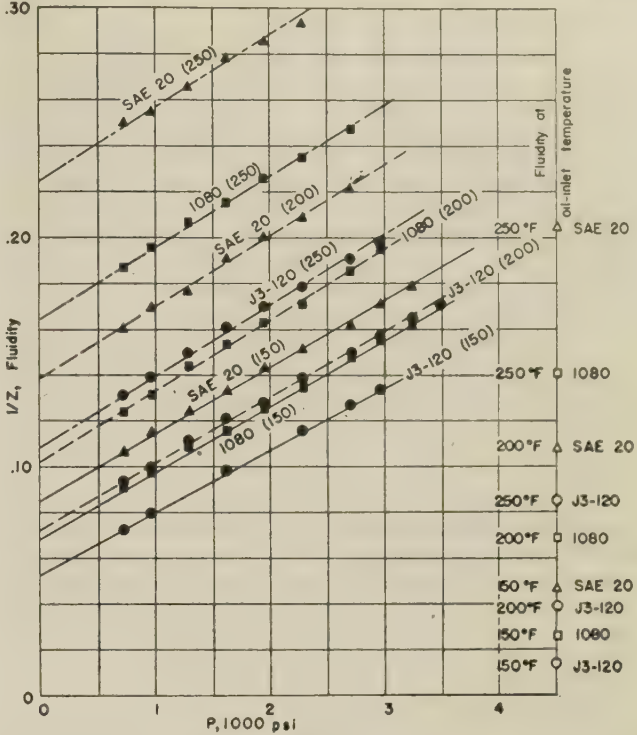


FIG. 11 DATA SHOWING EFFECTS OF VISCOSITY GRADE AND OIL-INLET TEMPERATURE ON FLUIDITY-PRESSURE CHARACTERISTICS WITH BEARING SET 1b
(2-in. \times 1 $\frac{1}{4}$ -in. copper-lead bearing, $L/D = 556$, 3000 rpm, 35 psi average oil pressure. Figures in parentheses indicate oil-inlet temperature, deg F.)

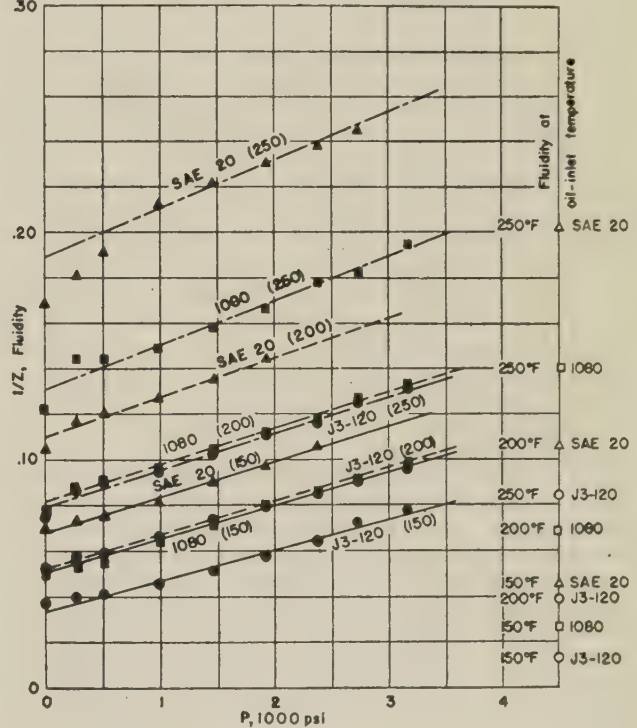


FIG. 13 DATA SHOWING EFFECTS OF VISCOSITY GRADE AND OIL-INLET TEMPERATURE ON FLUIDITY-PRESSURE CHARACTERISTICS WITH BEARING SET 3a
(1 $\frac{1}{4}$ -in. \times 1 $\frac{1}{4}$ -in. copper-lead bearings, $L/D = 1.002$, $D/C = 500$, 2080 rpm, 35 psi average oil pressure. Figures in parentheses indicate oil-inlet temperature, deg F.)

From the standpoint of hydrodynamics the friction increases with a decrease in clearance provided the viscosity remains the same in the two cases. From the standpoint of thermodynamics, however, a decrease in clearance decreases the oil flow, with a consequent increase in operating temperature and decrease in viscosity. This reduction in viscosity tends to decrease the friction. In these particular tests apparently the thermodynamic effect predominated under conditions of high load. The data for these bearings, when using other oils or when operating at other oil-inlet temperatures, indicate that this difference in slope between bearings of different D/C ratios is in general smaller with a decrease in viscosity grade or an increase in oil-inlet temperature.

A comparison of curve 3b ($D/C = 329$, $L/D = 1$) with curve 2c ($D/C = 338$, $L/D = 0.63$), indicates that the increase in friction with unit increase in load decreases slightly with an increase in length-diameter ratio. Tests with other oils or at other oil-inlet temperatures show relative differences of about the same order of magnitude.

The effect of size (diameter) on the increase in friction with unit increase in load is shown by a comparison of curves 1a with 2a and 1c with 2c in Fig. 9. With the large clearance-diameter ratio the curve for the small bearings is slightly steeper than that for the larger bearings. However, with the small clearance-diameter ratio, the trend is reversed. These differences tend to decrease with a decrease in viscosity or increase in oil-inlet temperature.

Most of the tests were made at approximately the same oil-feed pressure (averaging about 35 psi). Tests where the oil-feed pressure was changed were confined to operation at 2000 or 2080 rpm with the Navy Contract 1080 oil at 200 F oil-inlet temperature. Typical torque versus bearing-pressure curves, showing the effects of these changes, are given in Fig. 10. These indicate an increase in torque with increase in oil-feed pressure.

FLUIDITY DATA

Figs. 3, 4, and 5 show also that, when a given set of bearings is operating at a constant speed with a given oil, oil-inlet temperature, and oil-feed pressure, an increase in load produces a proportional increase in the fluidity of the oil at the bearing temperature. This straight-line relationship between fluidity and load also holds reasonably well in all the tests. This is of particular interest when consideration is given to the complex manner in which the viscosity of the oil functions in the operation of a journal bearing in a steady state of temperature distribution. The viscosity directly affects the bearing friction, and hence the rate of heat generation. It also affects the rate of oil flow, and hence the rate of heat dissipation by the oil. In this case its function is more complex. Primarily it is the property which produces a force resisting flow. It also affects the eccentricity between journal and bearing which in turn has an effect upon the rate of flow of the oil through the bearing.

The effects of viscosity and oil-inlet temperature upon the fluidity-pressure characteristics of some of the bearings tested are shown in Figs. 11, 12, and 13. Fluidity data are given for operation with all three oils at all three oil-inlet temperatures when testing set 1b at 3000 rpm, set 2c at 3050 rpm, and set 3a at 2080 rpm. The fluidity of the oils at the various oil-inlet temperatures is indicated along the right-hand border of these figures. A comparison of these data with the curves shows the influence of the fluidity of the oil at oil-inlet temperature upon the performance of the bearings. It will be noted also that the curves are roughly parallel, with a trend for a slight increase in slope with increase in oil-inlet temperature or decrease in viscosity.

* Analysis of the data for all sets of bearings indicates that, when operating with a given oil at a given oil-inlet temperature, the increase in fluidity with unit increase in bearing pressure (slope

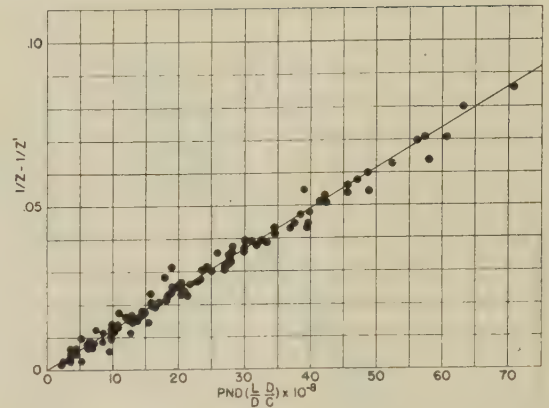


FIG. 14 DATA SHOWING PROPORTIONALITY OF INCREASE IN FLUIDITY WITH PRODUCT OF BEARING PRESSURE, JOURNAL SPEED, DIAMETER, LENGTH-DIAMETER RATIO, AND DIAMETER-CLEARANCE RATIO, FOR SAE 20 OIL AT 150 F OIL-INLET TEMPERATURE AND 36 PSI OIL PRESSURE

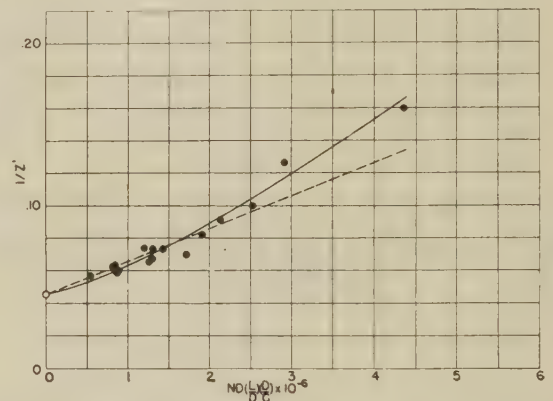


FIG. 15 DATA SHOWING RELATIONSHIP BETWEEN $1/Z'$ AND $ND (L/D) (D/C)$ FOR SAE 20 OIL AT 150 F OIL-INLET TEMPERATURE AND 36 PSI AVERAGE OIL PRESSURE

of the $1/Z$ versus P curve) is approximately proportional to the speed of the journal, the diameter of the journal, the length-diameter ratio, and the diameter-clearance ratio. This is shown in Fig. 14, where $1/Z - 1/Z'$ is plotted against $PND (L/D) (D/C)^5$ for the tests using the SAE 20 oil at 150 F oil-inlet temperature with all eight sets of bearings. The straight line in this figure represents the average slope for all the bearings.

There is also an indication that the values of the zero intercept, $1/Z'$, are a function of the variable $ND (L/D) (D/C)$. This is shown in Fig. 15 where the values of $1/Z'$, for lines of average slope for this same series of tests with the SAE 20 oil at 150 F oil-inlet temperature are plotted against this variable. The point at the left border of this figure, is the fluidity of the oil at the oil-inlet temperature. From this figure it will be noted that, for the range of $ND (L/D) (D/C)$ covered, the data may be represented roughly by the broken straight line or more nearly by the solid curved line.

Consideration of Figs. 13 and 14 indicates that, under the conditions covered, the fluidity of the oil in these bearings may be represented roughly by an empirical equation of the form

$$\frac{1}{Z} = \frac{1}{Z_0} + (aP + b) \left[ND \left(\frac{L}{D} \right) \left(\frac{D}{C} \right) \right] \dots \dots \dots [1]$$

⁵ This can be simplified by canceling D . For convenience, however, it is expressed in terms of the parameters L/D and D/C .

or more nearly by the equation

$$\frac{1}{Z} = \frac{1}{Z_0} + m \left[ND \left(\frac{L}{D} \right) \left(\frac{D}{C} \right) \right]^n + aP \left[ND \left(\frac{L}{D} \right) \left(\frac{D}{C} \right) \right] \quad [2]$$

Using $a = 1.224 \times 10^{-11}$ (slope of the straight line in Fig. 14), and $b = 2.01 \times 10^{-8}$ (slope of the straight line in Fig. 15), departures of the experimental points from the straight lines defined by Equation [1] are considered as differences in $1/Z$ and can be expressed as corresponding differences in bearing temperature. The greatest deviation found is equivalent to 16 F (at a bearing temperature of 243 F), and the average deviation is ± 3.8 F. With Equation [2], using $a = 1.224 \times 10^{-11}$, $m = 2.8 \times 10^{-10}$, and $n = 1.3$, the maximum deviation is equivalent to 8 F (at a bearing temperature of 205 F) with an average deviation of ± 2.9 F.

Similar data for the tests with the Navy Contract 1080 oil at 200 F oil-inlet temperature with seven sets of bearings (set 2a was not run with this oil) are given in Figs. 16 and 17. The constants for Equations [1] and [2] obtained from these figures are $a = 1.450 \times 10^{-11}$, $b = 1.59 \times 10^{-8}$, $m = 1.1 \times 10^{-11}$ and $n = 1.5$. Using these constants, the maximum deviation with Equation [1] is 17 F (at a bearing temperature of 268 F), and the average deviation is ± 4.6 F. The maximum deviation from Equation [2] is 9 F (at a bearing temperature of 264 F) with an average deviation of ± 2.9 F.

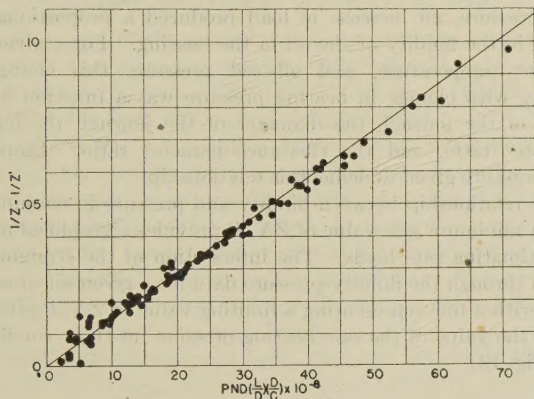


FIG. 16 DATA SHOWING PROPORTIONALITY OF INCREASE IN FLUIDITY WITH PRODUCT OF BEARING PRESSURE, JOURNAL SPEED, DIAMETER, LENGTH-DIAMETER RATIO, AND DIAMETER-CLEARANCE RATIO, FOR 1080 OIL AT 200 F OIL-INLET TEMPERATURE AND 34 PSI AVERAGE OIL PRESSURE

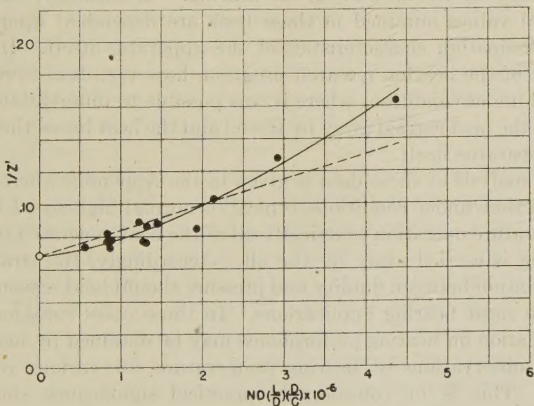


FIG. 17 DATA SHOWING RELATIONSHIP BETWEEN $1/Z'$ AND $ND(L/D)(D/C)$ FOR 1080 OIL AT 2000 F OIL-INLET TEMPERATURE AND 34 PSI AVERAGE OIL PRESSURE

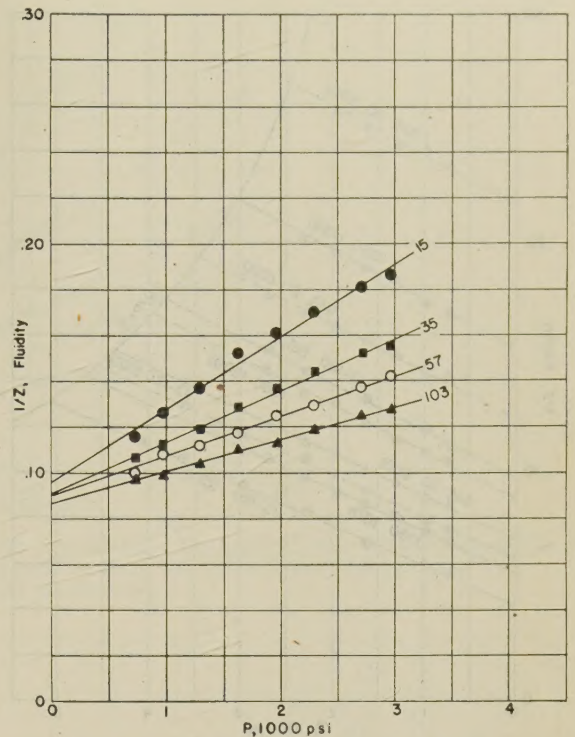


FIG. 18 FLUIDITY DATA SHOWING EFFECT OF OIL-FEED PRESSURE WITH BEARING SET 1b

(2-in. \times 1 1/4-in. copper-lead bearings, $L/D = 0.620$, $D/C = 556$, 2000 rpm, 1080 oil at 200 F oil-inlet temperature. Figures at ends of curves indicate oil-feed pressure, psi.)

These two cases used to indicate the general trend of the fluidity data cover an intermediate range of fluidities. Data for the other conditions have not been examined in detail. However, Figs. 11, 12, and 13, which are reasonably representative of all the data, show no marked differences in trend between the cases examined and operation with other oils and oil-inlet temperatures.

The equations just given are applicable to operation at one oil-feed pressure. Typical fluidity bearing-pressure data obtained at other oil-feed pressures for the same tests, described in Fig. 10, are given in Fig. 18. These indicate a decrease in fluidity with increase in oil-feed pressure.

GRAPHICAL SAFE-LOAD ESTIMATES

The straight-line relationship between fluidity and bearing pressure provides a graphical means for estimating safe loads. In this method the limit for safe operation is based upon a minimum value of ZN/P , as determined by the point of minimum f in the f versus ZN/P curve for a bearing. An example is given in Fig. 19. In this figure, the $1/Z$ versus P data are given for the operation of set 1b at 2000 rpm, when using all three oils at the three oil-inlet temperatures. The curved line at the right of the figure represents the constant value of $ZN/P = 3$, which experimental data have indicated as being representative of the limiting safe value for this set of bearings. The intersection of the straight line drawn through the $1/Z$ versus P data for any one set of conditions with the curved line representing the limiting value of ZN/P gives directly the value of the safe bearing pressure for those conditions.

Data pertinent to the question of extrapolation of the $1/Z$ versus P line to higher values of P are given in Fig. 20. These data were obtained in another series of tests where the loads were increased until unstable conditions were reached, using bearing

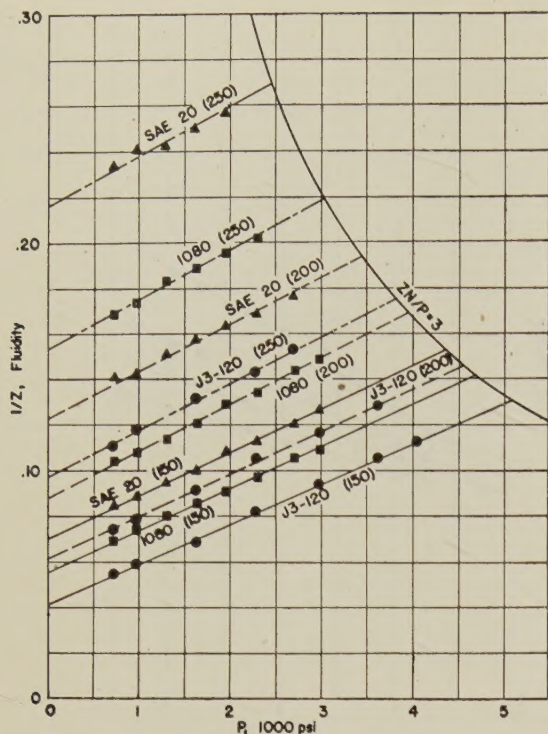


FIG. 19 DATA SHOWING EFFECT OF FLUIDITY ON SAFE OPERATING PRESSURES FOR BEARING SET 1b

(2-in. \times 1 1/4-in. copper-lead bearings, $L/D = 0.620$, $D/C = 556$, 2000 rpm, 35 psi average oil pressure. Figures in parentheses indicate oil-inlet temperature, deg F.)

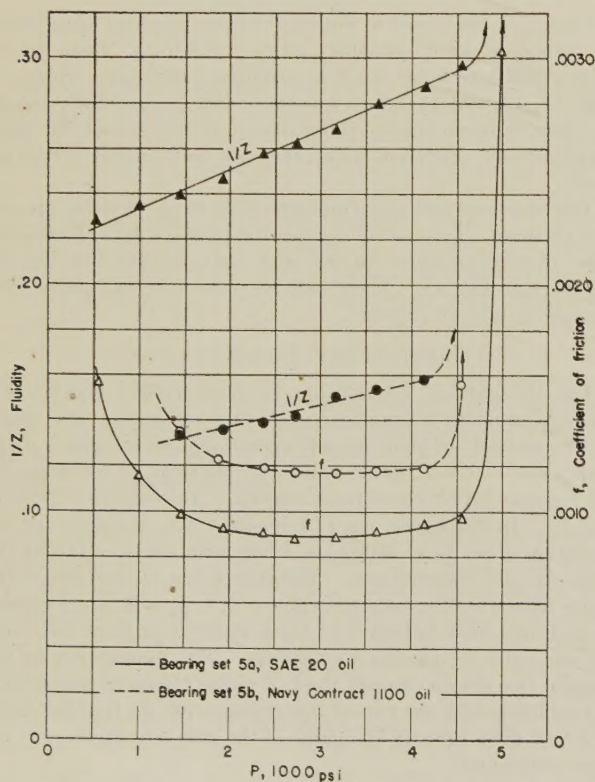


FIG. 20 FLUIDITY AND FRICTION DATA FOR LOW VALUES OF ZN/P AT 2000 RPM AND 250 F OIL-INLET TEMPERATURE

set 3a with the SAE 20 oil and bearing set 3b with a Navy Contract 1100 oil (not listed in viscosity table). In this figure, a comparison of the fluidity data and friction data indicates that under the conditions covered a straight line is reasonably representative of the $1/Z$ versus P data for loads up to the point of minimum friction.

SUMMARY OF RESULTS

Analysis was made of data obtained with the four-bearing friction machine in tests involving the operation of bearings of two diameters, two length-diameter ratios, three clearance-diameter ratios, using three oils of different viscosity at three oil-inlet temperatures and with two journal speeds. Particular attention was given to the resultant effects of the combined hydrodynamic and thermodynamic actions involved in bearing operation with forced-feed lubrication.

Analysis of the frictional-torque data for operation under load indicated that, when a bearing was operating at a constant speed using a given oil at constant oil-inlet temperature and oil-feed pressure, an increase in load produced an approximately proportional increase in torque. The increase in torque with unit increase in load was influenced chiefly by the viscosity of the oil, the oil-inlet temperature, the oil-feed pressure, the clearance-diameter ratio, and the length-diameter ratio.

The data also indicated that, when a bearing was operating at a constant speed with a given oil, oil-inlet temperature, and oil-feed pressure, an increase in load produced a proportional increase in the fluidity of the oil in the bearing. For a given oil, oil-inlet temperature, and oil-feed pressure, this change in fluidity with change in bearing pressure was a function of the speed of the journal, the diameter of the journal, the length-diameter ratio, and the clearance-diameter ratio. Empirical equations are given to define this relationship.

The relationship between fluidity and pressure in conjunction with a minimum safe value of ZN/P provides a graphical means for estimating safe loads. The intersection of the straight line drawn through the fluidity-pressure data for a given set of conditions with a line representing a limiting value of ZN/P gives directly the value of the safe bearing pressure for these conditions (see Fig. 19).

CONCLUSION

In considering these results it should be noted that the thermal behavior of a journal bearing operating with forced-feed lubrication is dependent upon a complex relation involving heat losses by radiation, conduction, and convection as well as the heat carried away by the oil leaving the bearing. Accordingly, the numerical values obtained in these tests are dependent upon the heat-dissipation characteristics of the apparatus itself. In the course of the bearing research program, however, data were obtained under conditions where it was possible to differentiate between the heat carried away by the oil and the heat losses through the apparatus itself.

An analysis of these data is given in the Appendix where it is shown that under conditions typical of normal high-speed high-temperature operation practically all of the heat generated in the bearing is carried away by the oil. Accordingly, the straight-line relation between fluidity and pressure should hold reasonably well in most bearing applications. In these cases considerable information on bearing performance may be obtained by making a few observations of bearing temperature for various conditions. This is of considerable practical significance since in many bearing installations friction measurements are not practicable.

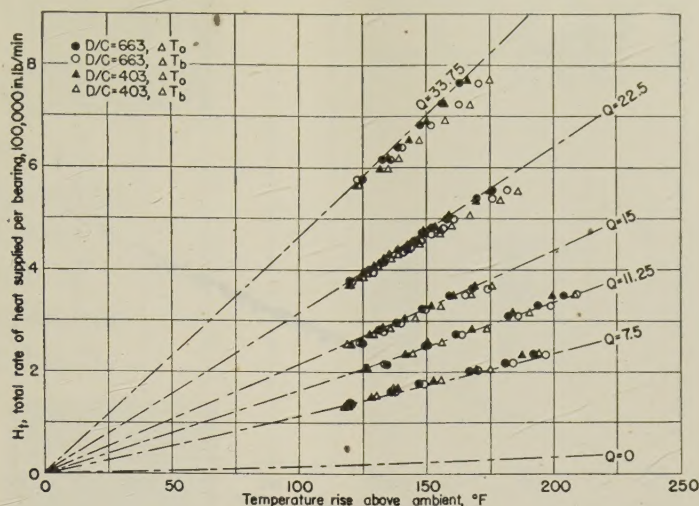


FIG. 21 TYPICAL DATA SHOWING THERMAL BEHAVIOR OF 2-IN. \times 1 $\frac{1}{4}$ -IN. COPPER-LEAD BEARINGS HAVING A CIRCUMFERENTIAL OIL GROOVE, WHEN OPERATING AT 2030 AND 3040 RPM, WITH SAE 20 OIL AT 200 F OIL-INLET TEMPERATURE (Values of Q are cubic inches per minute oil flow.)

Appendix

THERMAL BEHAVIOR AT CONSTANT OIL FLOW

In the course of the general program on bearing performance, some tests were made in which the oil flow was held at a constant rate during a test run. These tests were made at high ZN/P under a steady state of temperature distribution.

In the analysis of these data, consideration was given to the total heat supplied to the bearing. With the operating conditions such as were present in these tests, where the oil-inlet temperature was higher than the ambient temperature, the temperature rise of the bearings above the ambient was not only dependent upon the heat generated by shearing the oil in the bearing but also upon the heat delivered by the oil entering the bearing.

Typical data obtained in tests with two sets of bearings of different D/C ratios (663 and 403), operating at two speeds, various loads, and various rates of oil flow with an SAE 20 oil at 200 F oil-inlet temperature are given in Fig. 21. In this figure, H_t , the rate of heat supply to one bearing from both entering oil and bearing friction, is plotted against the temperature rise above the ambient. The solid points indicate the rise in temperature above the ambient of the oil leaving the bearings (ΔT_o), as determined by thermocouples located in the oil streams at the ends of each bearing, while the open points represent the temperature rise of the bearings (ΔT_b), as determined by thermocouples placed in the loaded sides of the bearing shells. Since these tests were made at a steady state of temperature distribution, the data are also indicative of the rate of heat dissipation.

Analysis of the data in this figure indicates that the relation between the rate of total heat supplied to the bearings and the temperature rise above the ambient is dependent chiefly upon factors affecting the rate of heat dissipation by the oil, namely, rate of oil flow, specific heat of the oil, and temperature rise of the oil. This is shown in Fig. 22. In this figure, H_t is plotted against Q , the rate of oil flow through one bearing, for various constant values of ΔT_o . The points shown in the figure were obtained from the data given in Fig. 21, interpolating where necessary. The data in Fig. 22 fall reasonably well on a series of straight lines, each of which represents the H_t versus Q relationship for a given value of ΔT_o . The slopes of these lines represent the heat-dissipation characteristics of the oil flowing through

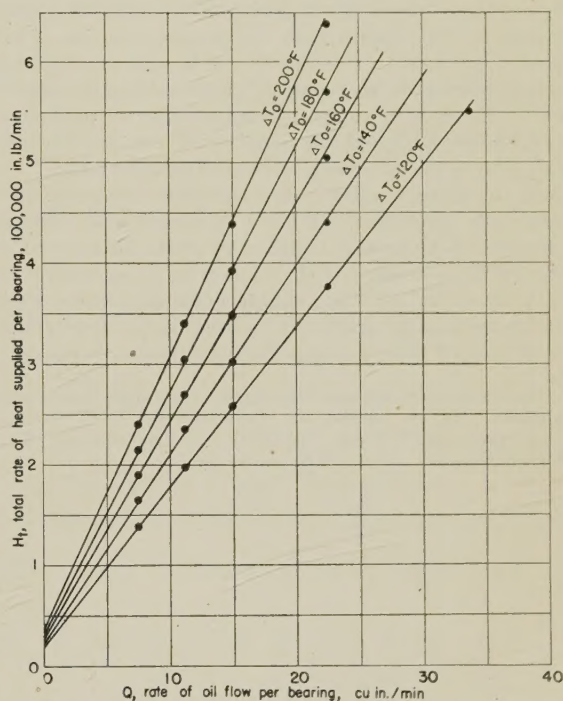


FIG. 22 H_t VERSUS Q CURVES FOR VARIOUS VALUES OF ΔT_o FOR 2-IN. \times 1 $\frac{1}{4}$ -IN. COPPER-LEAD BEARINGS HAVING A CIRCUMFERENTIAL OIL GROOVE, AT 2030 AND 3040 RPM, WITH SAE 20 OIL AT 200 F OIL-INLET TEMPERATURE

the bearing, while the intercepts at $Q = 0$ provide an indication of the rate of heat loss through the apparatus itself. This straight-line relationship between H_t and Q , together with the relative slopes of the lines for different values of ΔT_o , indicate the proportionality between the rate of heat removal by the oil and the rate of oil flow and the temperature rise.

The lines drawn in Fig. 21 represent the H_t versus ΔT_o relationship for the constant rates of oil flow indicated. The general location of the line $Q = 0$ is based on average values of the intercepts $Q = 0$, obtained from Fig. 22, together with similar analyses for other bearings tested. However, since the order of ac-

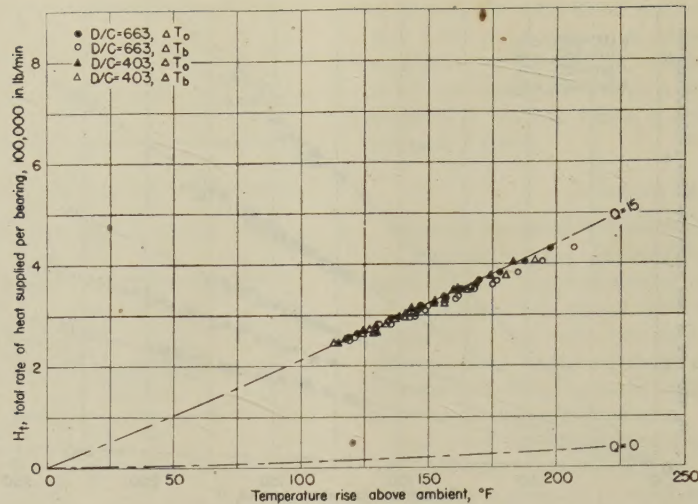


FIG. 23 TYPICAL DATA SHOWING THERMAL BEHAVIOR OF 2-IN. \times 1 $\frac{1}{4}$ -IN. COPPER-LEAD BEARINGS, HAVING OIL FED THROUGH HOLLOW SHAFT WITH TWO OIL HOLES PER BEARING WHEN OPERATING AT 1020, 2030, AND 3040 RPM, WITH SAE 20 OIL AT 200 F OIL-INLET TEMPERATURE AND 15 CU IN. PER MIN OIL FLOW*

curacy of the extrapolation is not such as to provide a sensitive indication of the exponent for the temperature rise, Lasche's value of 1.3 is used.⁶ This line $Q = 0$ is used as a base for the lines representing the H_t versus ΔT_o for the different values of Q . These are obtained by computations involving the product of the rate of oil flow indicated, the temperature rise, and the average specific heat of the oil over the given temperature range.

From this figure it will be noted that these computed lines are in reasonable agreement with all of the experimental data based upon values of the average temperature above the ambient of the oil leaving the bearings. In this connection the data cover operation at two speeds, with two clearance-diameter ratios, when using a circumferential groove in the bearing for distributing the oil. Data with other bearings having various arrangements of

holes and grooves show the same general trends. Using the hollow test shaft with two oil holes per bearing, data were obtained when operating with two clearance-diameter ratios, three speeds, and one rate of oil flow. These are given in Fig. 23 and it will be noted that they are in general agreement with Fig. 21.

Figs. 21 and 23 provide a comparison between the rate of heat removal by the oil and the rate of heat dissipation through the apparatus by radiation, conduction, and convection. It will be noted that the losses through the apparatus are relatively small even at the lower rates of oil flow. The rates of flow used in these tests cover a relatively low range. In some of the tests with the bearings having small C/D ratios, the fluidity-pressure data were obtained at low rates of oil flow, but the bulk of these data were obtained at the higher rates where the losses through the machine were practically negligible. This is probably also the case in most service applications of bearings operating with forced-feed lubrication.

* "Die Reibungsverhältnisse in Lagern mit höher Umfangsgeschwindigkeit," by O. Lasche, (a) *Zeitschrift des Vereines deutscher Ingenieure* vol. 46, 1902, pp. 1881-1890. (b) English trans., *Traction and Transmission*, vol. 6, 1903, pp. 33-64.